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A Stop-Hole Method for Marine and Offshore Structures

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Abstract

Stop holes have been widely used as temporary repair means in marine and offshore engineering to extend the service life of cracked structural components that cannot be immediately replaced. However, there is a lack of guidance with respect to the appropriate size of a stop hole that is usually determined based on the judgment of the attending surveyor/inspector. In the present paper, a method is proposed for determining the appropriate size of a stop hole for cracked marine and offshore structures. In addition to the traditional concerns over the size and nature of the crack, the proposed method incorporates high-cycle and low-cycle fatigue analyses into remain service life prediction taking into account both long-term and short term wave-induced loading. The effects of the return period of the potential severest sea condition, the crack length, and the environmental severity factor on the remaining service life of a stop hole are investigated.

Keywords: Stop-hole method, low-cycle fatigue, wave-induced loading, marine and offshore structures

1. Introduction

As a relatively inexpensive and easy application, the stop-hole method (Song and Shieh 2004; Murdani *et al.* 2008; Wu *et al.* 2010) has been widely used in the marine and offshore engineering to extend the service life of cracked structural components that cannot be moved in the near term to a repair facility and replaced. As shown in Fig.1, the purpose of this method is to transform a crack into a notch by drilling a hole in the vicinity of, or centred at, the crack tip. In this way, the crack propagation is temporarily stopped.

In marine and offshore engineering, current practice for determining the size of a stop hole is usually based on the judgment of the attending surveyor/inspector. There is a lack of guidance with respect to the appropriate size of the stop hole. Marine and offshore structures are normally exposed in the open sea and they are subjected to cyclic wave-induced loading. In addition to the traditional concerns over the size and nature of the crack, the proper determination of the size of a stop hole is not simple and needs to consider the fatigue behaviour of the already cracked structure within potential severe sea conditions before it is moved to a repair facility.

The wave-induced loading is usually described in two ways, using either long-term or short-term statistics. The cyclic stress magnitudes induced at a stop hole within most of sea conditions are normally far below the yield stress of material, and the time to initiate a new crack at a stop hole can be predicted by high-cycle fatigue analysis according to the long-term wave-induced loading. In severe sea conditions, the cycle stress magnitudes induced at the stop hole due to the short-term wave-induced loading may approach or exceed the yield stress of material and the time to initiate a new crack at the stop hole should be predicted by the low-cycle fatigue analysis using the short-term wave-induced loading generated within severe sea conditions. Therefore, to rationally determine the size of a stop hole, both high-cycle and low-cycle fatigue analyses for a stop hole should be performed in terms of the long-term and short-term wave-induced loading, respectively.

This paper proposes a rational method for determining the appropriate size of a stop hole, in which both high-cycle and low-cycle fatigue analyses are carried out according to the long-term and short-term wave-induced loading. The time to initiate a new crack at a stop hole subjected to the long-term wave-induced loading is predicted by characteristic S-N curves and the Palmgren-Miner's rule. The time to initiate a new crack of a stop hole within severe sea conditions are calculated by strain-life methods in terms of the short-term wave loading induced in severe sea conditions. The effects of the return period of the potential

severest sea condition, the crack length, and the environmental severity factor on the remaining service life of a stop hole are investigated.

2. Vertical wave-induced bending moment

In marine and offshore engineering, vertical wave-induced bending moment is usually described in two ways, using either short-term or long-term statistics. Long-term wave-induced loading is usually used for high-cycle fatigue analysis and wave-induced loading within a short-term duration corresponding to a severe sea condition is often utilized for low-cycle fatigue analysis.

2.1. Long-term probabilistic presentation

The probability density function of the amplitude of long-term vertical wave-induced loading x normally can be obtained by the weighted short-term probability density functions as follows (Ochi, 1978):

$$f_x(x) = \frac{\sum_i \sum_j \sum_k \sum_l n_* p_i p_j p_k p_l f_*(x)}{\sum_i \sum_j \sum_k \sum_l n_* p_i p_j p_k p_l} \quad (1)$$

where $f_*(.)$ is the short-term probability density function, n_* the average number of responses per unit time of short-term response, p_i the weighting factor for sea condition, p_j the weighting factor for wave spectrum, p_k the weighting factor for heading to the waves in a given sea, and p_l the weighting factor for speed in a given sea and heading.

In order to simplify the Eq.(1) for the practical engineering application, extensive studies (Jensen, 2001) were performed on the long-term vertical wave-induced loading and the results show the long-term probability density function of vertical wave-induced load $f_x()$ may be well approximated by a two-parameter Weibull distribution as

$$f_x(x) = \frac{k}{\lambda} \left(\frac{x}{\lambda} \right)^{k-1} \exp \left[- \left(\frac{x}{\lambda} \right)^k \right] \quad (2)$$

where k and λ are scale and shape parameters of the distribution, respectively. Therefore, the long-term vertical wave-induced loading acting on a stop hole is assumed herein to follow a two-parameter Weibull distribution.

2.2. Short-term probabilistic presentation

The amplitude of the vertical wave-induced loading within a short-term duration (typically several hours) corresponding to a steady sea state is usually considered to follow a Rayleigh distribution and the probability density function of the amplitude of short-term vertical wave-induced loading y is given by

$$f_y(y) = \frac{y}{s^2} \exp\left(-\frac{y^2}{2s^2}\right) \quad (3)$$

where s is the scale parameter of the distribution. As a result, the short-term vertical wave-induced loading acting on stop holes, i.e., vertical wave-induced loading during a storm, is assumed herein to follow a Rayleigh distribution.

2.3. Extreme value of vertical wave-induced loading

If the long-term vertical wave-induced loading x is assumed to follow a two-parameter Weibull distribution as shown in Eq.(2), in accordance with the extreme value theories, the extreme value x_e of vertical wave-induced loading within a given return period T could be assumed to follow a Gumbel distribution with a probability density function $f_e()$ and a cumulative density function $F_e()$ given as

$$\begin{aligned} f_e(x_e) &= \frac{1}{\sigma} \exp\left[-\frac{(x_e - u)}{\sigma}\right] \exp\left\{-\exp\left[-\frac{(x_e - u)}{\sigma}\right]\right\} \\ F_e(x_e) &= \exp\left\{-\exp\left[-\frac{(x_e - u)}{\sigma}\right]\right\} \end{aligned} \quad (4)$$

where u and σ are the location parameter and scale parameter of the distribution, respectively.

On the principle of the extreme value theories, u and σ were derived by Chen et al (2013) as

$$\begin{aligned} u &= \lambda (\ln n_w)^{\frac{1}{k}} \\ \sigma &= \frac{0.3679}{n_w [1 - 1/n_w]^{n_w - 1} f_x(u)} \end{aligned} \quad (5)$$

where n_w is the number of wave cycles during the given return period T .

3. High-cycle fatigue analysis

The time to initiate a new crack at a stop hole subjected to high-cycle fatigue is predicted herein by characteristic S-N curves and the Palmgren-Miner's rule. The two-segment S-N curves from ABS Rules (2015) are used for the prediction:

$$\begin{aligned} NS^{m_1} &= K_1 & \text{when } N \leq 10^7 \text{ cycles} \\ NS^{m_2} &= K_2 & \text{when } N > 10^7 \text{ cycles} \end{aligned} \quad (6)$$

where N is the number of loading cycles to failure, S is the stress range, m_1 and m_2 are the fatigue strength exponents, and K_1 and K_2 are the fatigue strength coefficients.

The high-cycle fatigue life is determined by the Palmgren-Miner's rule. When there are k different stress ranges S_i ($1 \leq i \leq k$) and each S_i contributes n_i cycles, then the fatigue failure is considered to occur when:

$$D = \sum_{i=1}^k \frac{n_i}{N_i} = 1 \quad (7)$$

where N_i is the number of loading cycles to failure under a constant stress range S_i and D is the fatigue damage.

By taking into account the two-segments S-N curves, the fatigue damage D can be reformulated as (Chen et al., 2011):

$$D = \Lambda \frac{nE(S^{m_1})}{K_1} \quad (8)$$

where n is the total number of stress cycles, that is:

$$n = \sum_{i=1}^k n_i \quad (9)$$

$E(S^{m_1})$ is the mean value of S^{m_1} and it is expressed as:

$$E(S^{m_1}) = \int_0^\infty S^{m_1} f(S) dS \quad (10)$$

If the long-term wave-induced loading acting on stop holes is assumed to follow a two-parameter Weibull distribution, it follows that

$$E(S^{m_1}) = A^{m_1} \left[\Gamma \left(1 + \frac{m_1}{\gamma} \right) \right] \quad (11)$$

where $\Gamma(\cdot)$ is the gamma function, and Λ is the bias factor defined as:

$$\Lambda = \frac{K_1}{K_2} A^{m_2 - m_1} \frac{\Gamma_0(z, y)}{\Gamma(x)} + \frac{\Gamma(z, x)}{\Gamma(x)} \quad (12)$$

where $\Gamma(z, x)$ and $\Gamma_0(z, y)$ are incomplete gamma functions given by:

$$\Gamma(z, x) = \int_z^\infty q^{x-1} e^{-q} dq \quad (13)$$

$$\Gamma_0(z, y) = \int_0^z q^{y-1} e^{-q} dq \quad (14)$$

where

$$x = m_1 / \gamma + 1 \quad (15)$$

$$y = m_2 / \gamma + 1 \quad (16)$$

$$z = \left[\frac{(K_1 / K_2)^{\frac{1}{m_1 - m_2}}}{A} \right]^\gamma \quad (17)$$

The fatigue life predicted by high-cycle fatigue analysis is then given by:

$$N_h = \frac{DK_1}{\Lambda E(S^{m_1})} \quad (18)$$

4. Low-cycle fatigue analysis

When the stress levels are sufficiently high for plastic deformation to occur in the low-cycle fatigue regime, the account in terms of stress is less useful and the cyclic stress-strain response and the material behaviour are better modeled under strain-controlled conditions (Bannantine et al., 1990).

4.1. Strain-life methods

In 1910, Basquin (1910) observed that the data of S-N curves may be formulated as:

$$\frac{\Delta\sigma}{2} = \sigma_f' (2N)^b \quad (19)$$

where $\frac{\Delta\sigma}{2}$ is the true stress amplitude, σ_f' is the fatigue strength coefficient, $2N$ are the reversals to failure,

and b is the fatigue strength exponent (Basquin's exponent).

In the 1950s, Coffin (1954) and Manson (1953) independently found that the data of plastic strain-life curves can also be plotted by a power law function as

$$\frac{\Delta\varepsilon_p}{2} = \varepsilon_f' (2N)^c \quad (20)$$

where $\frac{\Delta \varepsilon_p}{2}$ is the plastic strain amplitude, ε_f' is the fatigue ductility coefficient, c is the fatigue ductility exponent.

Based on Eqs.(19) and (20), the total strain can be written as

$$\frac{\Delta \varepsilon}{2} = \frac{\sigma_f'}{E} (2N)^b + \varepsilon_f' (2N)^c \quad (21)$$

where E is the Young's modulus of material. Eq.(21) is called the strain-life relation and it is the basis of the strain-life method.

To account for mean stress effects, Morrow (1968) modified the elastic term in Eq.(21) by taking into account the mean stress σ_0 and Eq.(21) is then reformulated as

$$\frac{\Delta \varepsilon}{2} = \frac{\sigma_f' - \sigma_0}{E} (2N)^b + \varepsilon_f' (2N)^c \quad (22)$$

Eq.(22) is called herein the “Morrow elastic” strain-life relationship.

Later, to maintain the independence of elastic-plastic strain ratio from mean stress, Manson and Halford (1981) modified Eq.(22) as

$$\frac{\Delta \varepsilon}{2} = \frac{\sigma_f' - \sigma_0}{E} (2N)^b + \varepsilon_f' \left(\frac{\sigma_f' - \sigma_0}{\sigma_f'} \right)^{c/b} (2N)^c \quad (23)$$

Eq.(23) is called herein as “Morrow elastic -plastic” strain-life relationship.

While Morrow (1968) and Manson and Halford (1981) introduced the mean stress σ_0 into Eq.(21), Smith et al. (1970) introduced the maximum stress $\sigma_{\max}$ into Eq.(21) and proposed another equation as

$$\frac{\Delta \varepsilon}{2} = \frac{(\sigma_f')^2}{E \sigma_{\max}} (2N)^{2b} + \frac{\sigma_f' \varepsilon_f'}{\sigma_{\max}} (2N)^{b+c} \quad (24)$$

4.2. Local stresses and strains

The use of the strain-life methods for low-cycle fatigue analysis needs knowledge of the local stresses and local strains.

4.2.1. Stress concentration factor

The theoretical stress concentration factor, K_t , is often used to relate the nominal stresses, σ_n , or, nominal strains, ε_n , to the local stresses σ and local strains ε . When the local stress levels are low, the K_t can be calculated as

$$K_t = \frac{\sigma}{\sigma_n} \quad (25)$$

Hirano (1950), Cox (1953), and Sobey (1963) found the “equivalent ellipse” concept useful for an opening such as two holes connected by a slit, as shown in Fig.2. If such a shape is enveloped by an ellipse, the theoretical stress concentration factor K_t values for the shape and the equivalent ellipse may be nearly the same. A crack with two stop holes at the tips of the crack is very similar to the shape of two holes connected by a slit. Therefore, the theoretical stress concentration factor K_t for a stop hole is given by the K_t of the “equivalent ellipse” as

$$K_t = 1 + 2\sqrt{2 + \frac{a}{r}} \quad (26)$$

where a is the half length of a crack and r is the radius of the stop hole.

4.2.2. Fatigue stress concentration factor

To take into account the effects of material type on the stress concentration factor K_t , the concept of *notch sensitivity* q was introduced as (Peterson, 1945):

$$q = \frac{K_f - 1}{K_t - 1} \quad (27)$$

where K_f is the fatigue stress concentration factor.

The values of q range from zero to unity. There are a number of analytical relationships developed for determining q . One of the most common analytical expressions was developed by Peterson (1959) as

$$q = \frac{1}{1 + \frac{b_1}{r}} \quad (28)$$

where b_1 is a material constant and r is the notch root radius.

4.2.3. Neuber/Ramberg-Osgood relationship

In terms of Neuber/Ramberg-Osgood relationship, the local stress and strain range $\Delta\sigma$ and $\Delta\varepsilon$ induced by the nominal stress range $\Delta\sigma_n$ can be calculated using the $\Delta\sigma\Delta\varepsilon$ loop equation:

$$\begin{aligned} K_f^2 \left[\Delta\sigma_n^2 + \frac{2E\Delta\sigma_n^{1+1/n'}}{(2K')^{1/n'}} \right] &= \Delta\sigma^2 + \frac{2E\Delta\sigma^{1+1/n'}}{(2K')^{1/n'}} \\ \Delta\varepsilon &= \frac{\Delta\sigma}{E} + 2 \left(\frac{\Delta\sigma}{2K'} \right)^{1/n'} \end{aligned} \quad (29)$$

where n' and K' are the coefficient and the exponent of the cyclic stress-strain curve.

4.3. Low-cycle fatigue damage

Similar to the calculation of high-cycle fatigue damage, in the present paper, low-cycle fatigue damage of a stop hole induced within a short-term severe sea condition is defined as:

$$D_l = \sum_{j=1}^{n_s} \frac{1}{N_j} \quad (30)$$

where n_s is the total number of wave-induced stress cycles within the short-term duration and N_j is the low-cycle fatigue life predicted by a strain-life method in terms of the j th wave-induced stress cycle within the short-term duration.

As discussed in Section 2.2, the amplitude of the wave-induced loading within a short-term duration is usually considered to follow a Rayleigh distribution. Therefore, the wave-induced stresses within the short-term duration are herein generated by Rayleigh pseudorandom numbers. If c is a random variable having the uniform distribution on $[0, 1]$, the amplitude of short-term wave-induced loading y is then given by

$$y = s\sqrt{-2\log c} \quad (31)$$

where s is the scale parameter of the Rayleigh distribution.

Wave-induced loading within a short-term duration, corresponding to a steady sea state, is often simulated using a Monte-Carlo approach. Therefore, D_l calculated by Eq.(30) in terms of the amplitudes of the short-term wave-induced loading y , generated by Eq.(31), is a random variable. In order to select a reasonable D_l for the remaining service life prediction of a stop hole, histograms of D_l within 5000 three-hour durations simulated corresponding to the potential severest sea condition in terms of three strain-life relationships: (a) “Morrow elastic”; (b) “Morrow elastic-plastic”; (c) “Smith-Watson-Topper”, are built as

shown in Fig.3. In terms of Fig.3, from the point of view of safety and as well as practical engineering applications, the D_l with a probability of exceedance of 1% should be sufficiently conservative. As a result, the maximum D_l within 100 three-hour durations simulated corresponding to the potential severest sea condition is selected for determining the remaining service life of a stop hole:

$$D_l = \max\{D_{l,1}, D_{l,2}, \dots, D_{l,100}\} \quad (32)$$

where $D_{l,i}$ is the low-cycle fatigue damage of a stop hole calculated within the i th simulated three-hour duration corresponding to the potential severest sea condition.

5. Remaining service life

Marine and offshore structures are normally exposed in the open sea. There is no guarantee to avoid the potential severest weather. From the point view of safety, it is necessary to account for the low-cycle fatigue damage D_l induced within the potential severest sea condition in the remaining service life prediction of a stop hole.

Based on the above conservative considerations and Eq.(18), the remaining service life of a stop hole is defined herein as:

$$N_f = \frac{(1 - D_l)K_1}{\Lambda E(S^{m_1})} \quad (33)$$

6. Parametric study

As shown in Fig.1, a four-inch, through-thickness, transverse crack in the main deck (16mm thick) of a tanker with two stop holes at the tips of the crack are utilized as an example to perform a parametric study and as well as to show how to rationally determine the size of a stop hole. The principal dimensions of the tanker is given in Table 1. The material properties of deck plate are listed in Table 2. The nominal long-term stress range on the stop hole is assumed to follow a Weibull distribution with a shape parameter of 0.859 and a scale parameter of 8.896 MPa. The nominal mean stress and maximum stress on the stop hole are 102.75 and 231.31 MPa. The relationships between the radius of stop hole r and the low-cycle fatigue damage D_l induced within a three-hour duration corresponding to the potential severest sea condition and the remaining service life of the stop hole N_f are investigated.

D_I is estimated by strain-life methods in terms of three different strain-life relationships, namely, “Morrow elastic” (Eq.22), “Morrow elastic -plastic” (Eq.23), and “Smith-Watson-Topper” (Eq.24) strain-life relationships. The effects of the return period of the potential severest sea state, the crack length, and the environmental severity factor on D_I and N_f are shown in Figs.4-9, respectively.

Figs.4-9 generally show that the D_I decreases but N_f increases with the increase of the radius of the stop hole r . This is because the increase of r generally contributes to the decrease of the stress concentration factor K_t at the stop hole so as to increase N_f . However, it must be noted that the increase of r in the meantime also increases the nominal stress at the stop hole because it loses materials after drilling stop holes. Thus, the size of the stop hole cannot be too large. In addition, Figs.4, 6, and 8 show that D_I estimated in terms of the “Morrow elastic-plastic” strain-life relationship is much higher than those estimated according to “Morrow elastic” and “Smith-Watson-Topper” strain-life relationships. This indicates that the prediction of the “Morrow elastic-plastic” strain-life relationship may be more conservative in comparison to those of “Morrow elastic” and “Smith-Watson-Topper” strain-life relationships.

6.1. Return period of the potential severest sea condition

Return period of the potential severest sea condition selected for D_I estimation is an important factor to the selection of an appropriate size of a stop hole. Normally, the longer return period is considered, the severer sea condition is chosen for the potential severest sea condition for D_I estimation and the larger size of a stop hole may be chosen. The effects of the return period of the potential severest sea condition on D_I and N_f are shown in Figs.4 and 5 and the return periods of 1 year, 25 years and 100 years are selected for the parametric study.

Figs. 4 and 5 show that D_I significantly increases and N_f decrease with the increase of the return period of the potential severest sea condition selected for D_I estimation. This is because the longer return period is selected and the severer sea condition is chosen for the potential severest sea condition for D_I estimation. Thus D_I is significantly increased with the increase of the return period of the potential severest sea condition, which leads to the decrease of N_f .

It is important to note that the selected return period of the potential severest sea condition for D_I estimation should not be too short from the point of view of safety. It is recommended in the present paper that the return period of the potential severest sea condition chosen for D_I estimation should not be less than

one year. However, the selected return period should also not be too long and it is not necessarily more than the design life of the structure.

6.2. Crack length

Crack length is a key parameter affecting the radius of the stop hole r . As seen from Eq.(26), the stress concentration factor K_t is significantly influenced by the crack length $2a$. The effects of the crack length on D_I and N_f are shown in Figs.6 and 7, in which the crack length of 50.8mm, 101.6mm, 152.4mm, and 203.2mm are selected for the parametric study and the return period of the potential severest sea condition for D_I estimation is chosen to be one year.

As expected, Figs.6 and 7 show D_I dramatically increases and N_f significantly decreases with the increase of the crack length $2a$. It is because the increase of the crack length $2a$ leads to a significant increase of K_t at the stop hole and thus a significant increase of D_I and a dramatic decrease of N_f . Consequently, special attentions must be paid to the measurement of the crack length $2a$ before the size of a stop hole is determined.

6.3. Environmental severity factor

Ideally, the selected loads for fatigue analysis are to reflect the season and the routes a structure may sail. It is preferable to use the measured fatigue loading though such information is not always available in most cases. A practical alternative is to use design loads, which are defined and specified in design rules such as *ABS Rules* (2015). If the season and the intended routes are to be considered, the design loads need to be adjusted for the planned season and route. This can be achieved by calculating the environmental severity factors applying the service conditions that represent the effect of wave conditions of specific sites or specified routes on the fatigue loading.

The environmental severity factor β is often defined as a severity measure of the intended environment relative to the based environment in terms of extreme loads. It is given by

$$\beta = \frac{L_s}{L_n} \quad (33)$$

where L_s is the most probable extreme value based on site-specific environment with design return period for the dynamic load parameters. L_n is the most probable extreme value based on unrestricted North Atlantic environment with design return period for the dynamic load parameters.

The effects of environmental severity factor β on D_l and N_f are shown in Figs.8 and 9, in which the β of 0.7, 0.8, 0.9, and 1.0 are selected for the parametric study and the return period of the potential severest sea condition for D_l estimation is chosen to be one year. It can be seen from Figs.8 and 9 that the increase of the environmental severity factor β leads to a dramatic increase of D_l and as well as a significant decrease of N_f , which means that it is important for D_l estimation to apply the potential severest sea condition that represents specific sites or specified routes the cracked structure may operate/sail.

7. Minimum allowable size of stop hole

In this section, a practical implementation procedure is developed for determining the minimum allowable size of a stop hole, in which an appropriate return period of the potential severest sea condition for D_l estimation is suggested and rational acceptance criteria are proposed.

7.1. Potential severest sea condition

Severe sea conditions are the sea conditions developed in harsh weather sea environments. The potential severest sea condition of the design return period is typically used as the design base for marine and offshore structures. For instance, ship structural designs are normally based on the severest sea condition for the North Atlantic in a 20 ~ 25 year return period, while structural designs of offshore structures are normally based on the potential severest sea condition for the specified site for a 100-year return period.

As seen from section 6.1, the longer return period is considered, the severer sea condition is selected for the potential severest sea condition for D_l estimation and the larger size of a stop hole should be applied. From the point view of practical engineering applications, the return period of the potential severest sea condition selected for D_l estimation is suggested herein to be at least one year.

7.2. Acceptance criteria

The following acceptance criteria are proposed for determining the minimum allowable size of a stop hole, which may provide a comprehensive basis for the decision making process:

- The time to initiate a new crack at a stop hole predicted by Eq. (33), N_f , using high cycle and low cycle fatigue analysis should be equal to or greater than ($\geq$) the length of the intended voyages before the cracked structure enters a repair facility;
- The short-term duration of the potential severest sea condition for D_I estimation using low-cycle fatigue analysis is chosen to be 3 hours.
- The return period of the potential severest sea condition selected for D_I estimation should be equal to or greater than ($\geq$) 1 year.

7.3. Implementation procedure

Based on the acceptance criteria, a practical implementation procedure for determining the minimum allowable size of a stop hole is then proposed as:

- Calculate the fatigue stress concentration factor of the stop hole K_f with the use of the measured crack size and an assumed stop hole size.
- Establish the long-term distribution of the stress range on the stop hole according to the intended voyages before the cracked structure enters a repair facility.
- Determine the potential severest sea condition during the intended voyage before the cracked structure enters a repair facility.
- Predict the time to initiate a new crack at the stop hole N_f using Eq.(32) based on the high cycle and low cycle fatigue analysis.
- Plot the curve $r - N_f$, as shown in Fig.10, the minimum allowable size of the stop hole is then determined from the plot.

7.4. Examples

For the example utilized in the parametric study, if the length of the intended voyages before the cracked structure enters a repair facility is one year. As seen from Fig.10, the minimum allowable radius of the stop hole are given by 6.9 mm, 7.1 mm, and 7.6 mm in terms of “Morrow elastic” (Eq.22), “Morrow elastic - plastic” (Eq.23), and “Smith-Watson-Topper” (Eq.24) strain-life relationships. So, from the point of view of safety, the minimum allowable radius of the stop hole is then selected to be 7.6 mm.

8. Conclusions

A method is developed for helping surveyors/inspectors determine an appropriate size of a stop hole for temporarily repairing cracked marine and offshore structures. The proposed method incorporates high-cycle and low-cycle fatigue analyses into remain service life prediction of a stop hole taking into account both long-term and short term wave-induced loading. A parametric study is performed and the results show that a rational choice of stop hole size not only needs to consider the size and nature of the crack, but also needs to account for additional conservatism, *i.e.*, the choice of the return period of the potential severest sea condition selected for low-cycle fatigue damage estimation of a stop hole induced within a short-term duration, and as well as the environmental severity factor. In addition, the crack length is also an important factor on the selection of the appropriate size of a stop hole.

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Table 1. Principal dimensions of the tanker

Length overall L (m)	238
Breadth B (m)	32.3
Depth D (m)	18.3
Draft d (m)	11.7

Table 2. Material properties of deck plate

Symbol	Value
σ'_f	805.82 MPa
b	-0.1010
ε'_f	0.3609
c	-0.5540

Figure 1 A through-thickness crack with two stop holes in the main deck of a tanker

Figure 2 The “equivalent ellipse” concept

Figure 3. Histogram of low-cycle fatigue damage D_l induced within 5000 three-hour durations simulated corresponding to the potential severest sea condition in terms of three strain-life relationships: (a) “Morrow elastic”; (b) “Morrow elastic-plastic”; (c) “Smith-Watson-Topper”.

(a)

(b)

(c)

Figure 4. Relationships between the radius of stop hole r and the low-cycle fatigue damage D_I induced within the three-hour duration simulated corresponding to the potential severest sea condition with different return periods (1 year, 25 years, and 100 years) in terms of three strain-life relationships: (a) “Morrow elastic”; (b) “Morrow elastic-plastic”; (c) “Smith-Watson-Topper”.

(a)

(b)

(c)

Figure 5. Relationships between the radius of stop hole r and the remaining service life of the stop hole N_f , in which D_I is estimated in terms of strain-life relationships: (a) "Morrow elastic"; (b) "Morrow elastic-plastic"; (c) "Smith-Watson-Topper" and different return periods (1 year, 25 years, and 100 years) of the potential severest sea condition are considered.

(a)

(b)

(c)

Figure 6. Relationships between the radius of stop hole r and the low-cycle fatigue damage D_I induced within the three-hour duration simulated corresponding to the potential severest sea condition in terms of three strain-life relationships: (a) “Morrow elastic”; (b) “Morrow elastic -plastic”; (c) “Smith-Watson-Topper”, in which different crack lengths $2a$ (50.8mm, 101.6mm, 152.4mm, and 203.2mm) are considered.

(a)

(b)

(c)

Figure 7. Relationships between the radius of stop hole r and the remaining service life of the stop hole N_f , in which D_I is estimated in terms of strain-life relationships: (a) “Morrow elastic”; (b) “Morrow elastic -plastic”; (c) “Smith-Watson-Topper” and different crack lengths $2a$ (50.8mm, 101.6mm, 152.4mm, and 203.2mm) are considered.

(a)

(b)

(c)

Figure 8. Relationships between the radius of stop hole r and the low-cycle fatigue damage D_I induced within the three-hour duration simulated corresponding to the potential severest sea condition in terms of three strain-life relationships: (a) “Morrow elastic”; (b) “Morrow elastic-plastic”; (c) “Smith-Watson-Topper”, in which different environmental severity factor β (0.7, 0.8, 0.9, and 1.0) are considered.

Figure 9. Relationships between the radius of stop hole r and the remaining service life of the stop hole N_f , in which D_I is estimated in terms of strain-life relationships: (a) “Morrow elastic”; (b) “Morrow elastic-plastic”; (c) “Smith-Watson-Topper” and different environmental severity factor β (0.7, 0.8, 0.9, and 1.0) are considered.

Figure 10. Relationships between the radius of stop hole r and the remaining service life of the stop hole N_f , in which D_I is estimated in terms of strain-life relationships of “Morrow elastic”; “Morrow elastic -plastic”; and “Smith-Watson-Topper”.