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Fatigue Analysis Method for LNG Membrane Tank Details

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ABSTRACT

This paper gives a description of a fatigue analysis method based on the S-N curve and crack propagation method.

The presentation of the method is based on an application to a practical example, taken from Shipbuilding, i.e., the analysis of details of membrane LNG carriers of the Gaz Transport system type.

The paper, after a general presentation of the method, gives an analysis of the loads and computer calculations performed, to obtain stress levels.

It also presents the way that S-N curves and the necessary parameters for cumulative damage have been defined and give the results of Miner sum for two particular membrane details.

1 INTRODUCTION

The last decade saw a large development of the use of computer calculations in ship design studies. Finite element computer programmes are now widely used for hull structure design and scantlings. This possibility is also used for elements which are linked to the structure.

Calculations are nowadays used, for every type of ship and obviously for the most sophisticated, as for example LNG carriers. For this kind of ship, computations are done for hull, but also to design the containment system which is supported or linked to the hull structure. Computations allow the determination of loads or displacements transfered from the hull to the tanks, but also proper loads on tanks and stresses in tank elements.

Mainly, finite element methods are used ; they permit stresses in elements and also stresses in very localized areas such as element connections to be calculated. One can find in the reference literature, many papers dealing with such calculations. We shall just mention two of them as examples, dealing with spherical tanks system and membrane tank systems { 1, 2 }.

Such calculations allow the determination of stress levels, their global or local origin, whether they are static or dynamic, or if they are due to waves.

The stresses induced by the ship behaviour in waves are dynamic, i.e., cyclic stresses, and therefore induce a cyclic fatigue in elements and welds.

To define scantlings, when the loads are static, the calculated stresses are generally compared to material characteristics, such as yield and ultimate strength. For cyclic stresses, it is necessary to have a comparison with the cyclic fatigue capability of the material or welds. Therefore the design verification implies the use of fatigue analysis or crack propagation methods.

The development in fatigue and crack propagation calculation methods during the last several years allows the use of precise data and methods.

This paper presents a method which is used for detailed analysis of membrane tank LNG carriers, with particular examples from the Invar Gaz Transport system. Comments are given on the choice of methods and is followed by a general description. Then, practical application on two containment system details is developed.

2 METHOD PRESENTATION

When trying to select a procedure, two methods were available, based on S-N curves or on crack propagation.

The S-N curve method is more classical than crack propagation, the development of which is much more recent. The first idea would be to use the more recent and sophisticated method, but a detailed analysis of the method shows us a number of necessary assumptions that is too large. We must remember that in a membrane containment system, all elements are thin, the invar membrane of Gaz Transport system is for example 0.7 mm. The life span is defined as the necessary time under ships loads to obtain a leakage, i.e. to obtain a crack propagating through the membrane thickness.

Propagation laws of limited cracks in 2 D or 3 D remain difficult to establish and requires experimental data which are not available for such thin plates.

To use a crack propagation method, it is also necessary to define the size of the initial crack and to verify the initiation period, i.e. the time necessary under cyclic load, to create a sharp crack. This is small, with respect to propagation period, i.e. the time necessary under cyclic load, to propagate a sharp crack until perforation of the element.

In practice, the determination of the initial crack length and verification of the initiation period length is made by means of laboratory tests, the same tests that are used to determine S-N curve.

Taking into account the above, we decided to use the classical S-N curve method , with the help of crack propagation theory to determine for example the fatigue limit under random loads or the influence of static stresses on fatigue life, of a sample submitted to cyclic loads.

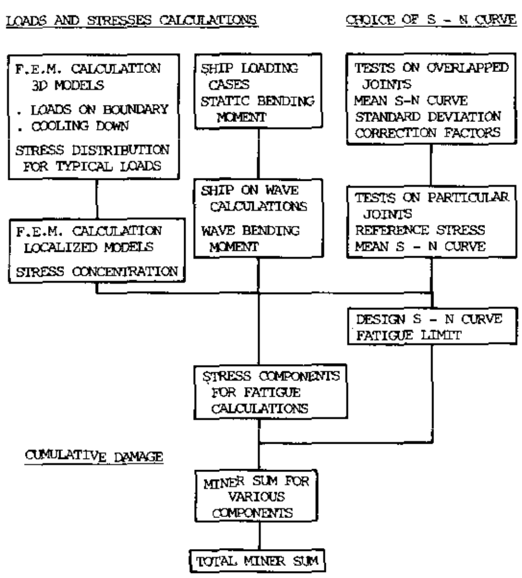

Fig. 1. Flow chart of the fatigue calculation

Figure 1 shows the flow chart of the method used. It can be summarized by three main steps :

- load and stresses calculations,
- choice of a design S-N curve
- cumulative damage calculation.

The first step is the calculation of loads which are applied to the membrane, and of the stresses in the most loaded areas. One can select three typical loadings, the loads which are transmitted from the hull to the membrane, the loads which are due to the tank cooling down from a global point of view, and the local thermal stresses originating when two different materials are linked together.

The hull transmitted loads implicate hull stress calculations.

Two types of loads exist :

- the static component due to the loading of the ship,
- the dynamic component due to waves.

The static component is computed by classical naval architecture programmes { 3 }.

The dynamic component requires the calculation of ship behaviour in regular waves, in irregular waves and then the development of a long term statistical analysis { 4, 5 }. In this way, it is possible to calculate the boundary loads of the tanks. Finite element calculations applied to tank components permit the determination of the resulting stresses in the most loaded areas.

The same finite element calculation permits the determination of the global effect of the cooling down between ambient temperature and LNG temperature (- 160°C)

In some cases, local finite element calculation might be necessary to determine local thermal stresses due to different contraction properties of different materials.

In order to be able to analyse the stress computation results, one must define S-N curves. S-N curves are determined from laboratory tests on elements or joints as similar as possible to the real ones. When the tested samples are somewhat different, it could be necessary to adapt the results to the real cases or to the available stress figures from the finite element calculations.

The laboratory tests are conducted for a particular static level and therefore a correction between test conditions and real conditions on the ship must be done. Two existing correction methods are available. One is the Goodman-Smith, the other is the K factor. The Goodman-Smith method was mainly established for basic material samples and requires the use of the material characteristics, yield and ultimate strength, associated with computed stresses.
The K factor method is based on crack propagation theory and only uses the minimum to maximum cyclic stress ratio R.

The studied details on membranes and weld joints which present initial defects and for which the local stresses in the areas of these defects, i.e., crack propagation areas, are not well known.

Therefore the K factor method was prefered as it does not require the calculation of the local stress concentration and the correcting factor can be easily determined by larboratory tests.

Then two cyclic loads which induce a stress range $\Delta\sigma_1$ and $\Delta\sigma_2$ with a minimum to maximum stress Ratio R_1 and R_2 are equivalent, from fatigue life point of view, if :

$$K_1 \quad (R_1) \, \Delta\sigma_1 = K_2 \quad (R_2) \, \Delta\sigma_2$$

Finally, the S-N curve must also take into account the scatter of the laboratory test results. The test results are analysed in terms of a mean deviation. It is assumed that the result scatter follows a normal law distribution (Figure 2) and it is possible to define a lower limit with a given confidence estimation level and a given probability limit to have no risk of failure (Figure 3). This curve will be the design curve.

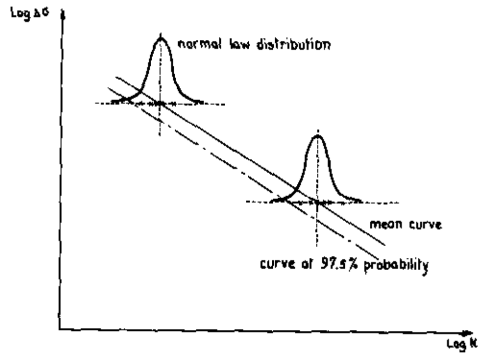

Fig. 2 Scheme of S-N curve with experimental scattering

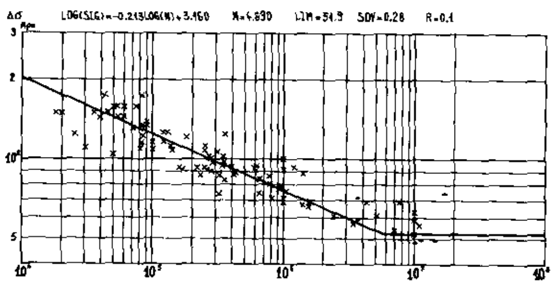

Fig. 3 Experimental results of fatigue tests on invar/invar lapped welds at room temperature. Stress range expressed in mean stress in the weld thickness, R = 0.1

Following these two studies for a given correction, for example, it is now possible to calculate the following stresses :

- static stress σ_{st} due to ship loading,
- thermal stress σ_{gth} due to global tank behaviour during the cooling down between ambient temperature and service temperature,
- local thermal stress σ_{lth} when two different materials are welded together,
- maximum wave stress σ_w at 10^{-8} probability level as required by I.M.C.O.

The wave stress is random and a long term distribution curve can be defined as it shall be seen later. Such a distribution can be represented by a stair curve (α_i , n_i) as shown in Figure 4.

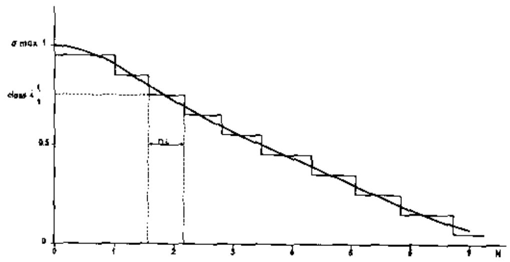

Fig. 4. Scheme of a long term stress distribution and step by step representation for Miner sum calculation

The S-N curve can be expressed by the following formula :

$$(\Delta\sigma)^m N = C \qquad \text{for a given } R_s \qquad (1)$$

where :

$\Delta\sigma$ = stress range

N = number of cycles to failure

m, C = experimental constants

R_s = minimum to maximum stress ratio.

To calculate the cumulative damage the Miner sum is used. The sum is expressed by :

$$\Sigma \frac{n_i}{N_i}$$

where :

n_i = number of cycles with stress range $\Delta\sigma$

N_i = number of cycles of the S-N curve at $\Delta\sigma$

Two types of cyclic stresses can be considered.

The first one is the cycling of thermal stresses between ambient temperature and operational temperature.

In this case, the maximum stress is equal to the service condition stress and the minimum stress is equal to O.

The range is :

$$\Delta\sigma = \sigma_{gth} + \sigma_{1th} \quad (2)$$

This range is constant for each cycle and the number of cycle n_1 is equal to the number of tank surveys during the ship life. The value of R is O. Therefore, if the S-N curve has been established for a value R which is different, the calculation of N_1 requires a correction of $\Delta\sigma$ by multiplication with a factor K.

The crack propagation theory shows that K can be expressed as follows { 6, 7 } :

$$K = (1 - a\ R)^{1/3} \quad (3)$$

where :

a is a constant which can be determined by tests.

The Miner sum component is therefore equal to :

$$\frac{n_1}{N_1} = \frac{n_1 (K\ \Delta\sigma)^m}{C} \quad (4)$$

For wave stress, which is the second cyclic type, it is necessary to consider the total range 2 σ_w, without taking into account any possibility of compressive values. This rule is necessary in order to take into account the existing residual stresses after welding. Essentially it has been found that in complex welded structures, when submitted to cycling loads, compressive stresses are as damaging as tensile stresses {8 }.

The long term statistical distribution of the wave stresses, is represented by a step by step curve as shown in Figure 4. A step level (i) is defined by the stress level to maximum stress at 10^{-8} probability ratio (α_i).

Now, for the wave stresses, at step i , a range is found :

$$\Delta\sigma = 2\ \alpha_i\ \sigma_w \quad (5)$$

with the associated static component :

$$\sigma_s = \sigma_{st} + \sigma_{gth} + \sigma_{1th} \quad (6)$$

Therefore one can calculate the correcting factor K_i by :

$$K_i = \left(\frac{1 - a\ R_i}{1 - a\ R_s} \right)^{-1/3} \quad (7)$$

where :

R_i is the R value at step (i)

$$R_i = \frac{\alpha_i \sigma_w - \sigma_s}{\alpha_i \sigma_w + \sigma_s} \quad (8)$$

and R_s is the R value of the S-N curve used.

The step by step curve, at step (i) defines a number of cycles n_i. The stress level $K_i\ \Delta\sigma$ on the S-N curve leads to a number of cycles N_i .

Therefore, the Miner sum components become

$$\Sigma \frac{n_i}{N_i} = \frac{\sigma_w^{\ m}}{C} \Sigma_1^M \ (\alpha_i\ K_i)^m\ n_i \quad (9)$$

where :

M is defined by the fatigue limit of the S-N curve.

But this limit, $\Delta\ \sigma_1$, is not the experimental one.

If one considers the life of a welded joint for example, the experimental fatigue limit corresponds to the level $\Delta\sigma_1$, below which the initial defect crack a_0 does not propagate.

Now, during cumulative damage on the ship, this initial crack of length a_0 , will propagate under the various stress levels. At a given time, it will be a, and that time, it can be increased by the level $\Delta\sigma_1$. This value is, therefore, no longer the fatigue limit. It is then necessary for assessment of cumulative damage, to determine another fatigue limit of the S-N curve. A practical example will be presented later.

3 LOAD AND STRESS CALCULATIONS

In order to present the method, the case of the invar membrane Gaz Transport system for LNG carriers is considered.

A global view of the tank disposition is given in Figure 5.

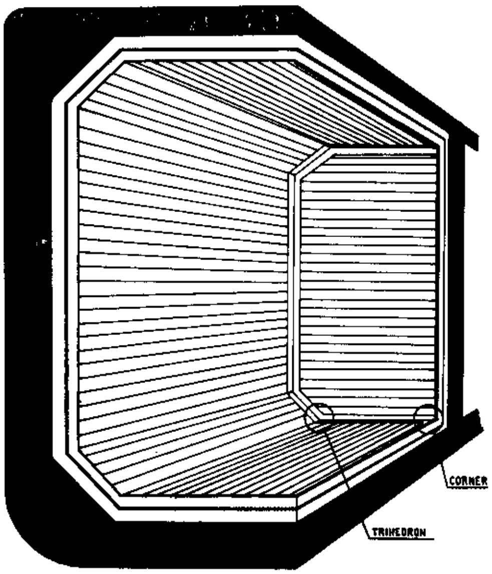

Fig. 5 General view of a quater of membrane LNG tank of Gaz Transport system type

The system arrangement is such that hull longitudinal stresses are transmitted to the flat invar membrane. Therefore itis first necessary to compute the hull longitudinal stresses which have two components,

. one static,

. one cyclic,

due to wave bending moments.

The hull stress at the level of the double hull will induce a uniform stress field in the membrane at a level which is equal to the hull stress level multiplied by the ratio of the Youngs modulus of invar to that of steel.

This stress field is created by the relative displacement of the end of transverse bulkhead of the tanks.

The static stress is calculated during study of the loading cases for the ship by classical methods {3} which are beyond the scope of this paper.

For the cyclic wave component stress determination, the classical method is used {4, 5} , of which an outline follows.

Ship motions in regular waves are calculated for various wave lengths, in order to be able to define the transfer function of the bending moment, at the midship section. The most commonly used method for ships is the strip theory with some improvements for hydrodynamic coefficient calculations{9 }. At the same time, the transfer functions in other sections are calculated to allow the definition of the longitudinal bending moment distribution.

Next the short term distribution of this bending moment for irregular sea-states is calculated. This distribution is defined by the spectral energy of the response bending moment and the coefficient of the corresponding Rayleigh distribution {4, 5} . This calculation is done for various sea headings, normally four angles from head sea to following sea, and for various sea-states defined by the mean period and a significant wave height equal to unity. The periods are taken from observation tables. The North Atlantic observation or the World Wide Tables can be used {10, 11} .

By combination of the results for each angle and sea-state, and taking the significant wave height distribution it is possible to calculate the long term distribution of the bending moments during the ship life {4, 5}.

Using the computer programmes, which have been developed by BUREAU VERITAS, such a curve was defined for the 130,000 m^3 class LNG tankers. The curve is given in Figure 6.

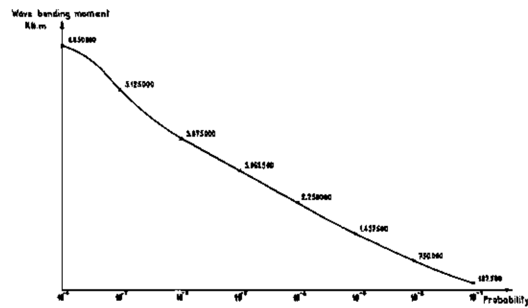

Fig. 6 Long term distribution of the midship wave bending moment of a 130,000 m^3 membrane LNG carrier

Taking into account a life time of 10^8 cycles, as given by I.M.C.O. regulations, one can define the curve of Figure 4.

As an example illustrating stress calculation, two details of the system are considered, the locations of which are given in Figure 5.

For the corner piece, which is called a trihedron, it has been possible to make a finite element model, which allows the calculation of the local stresses just at the edge of the weld joints. These joints are lapped joints between plates. The corner piece includes a stainless steel piece and invar plates with different characteristics. The model is given in Figure 8.

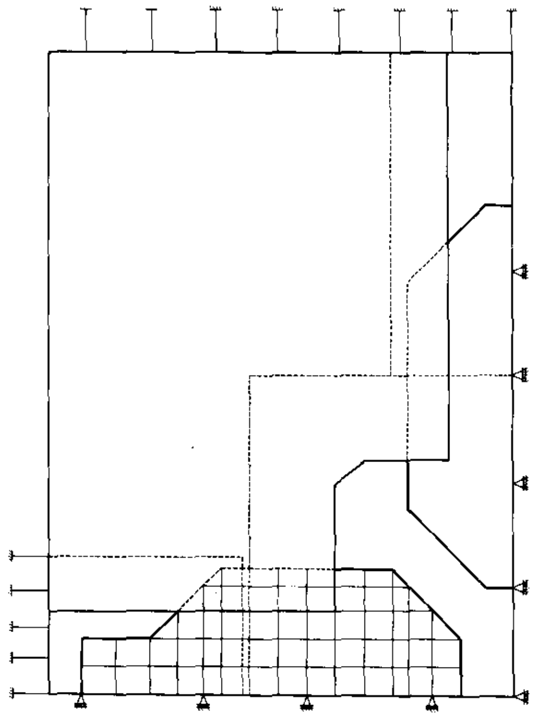

Fig. 8 Finite element model of the trihedron connection analysis with the mesh limited to one strap to allow better visibility.
This connection is refered in Figure 5.

This model was loaded by a forced displacement of the boundaries in center tank direction. This loading case allows one by scaling, to calculate stresses from hull stresses. The model, then clamped, was cooled down, allowing calculation of the local and global thermal stresses.

To analyse the results, Von Mises stresses were used.

It has been verified that the differences obtained with maximum shearing criteria, are not significant.

For the presented disposition, the most loaded point leads to the following stresses :

- wave stress at 10^{-8} probability : 41.7 MPa (6.0 Ksi)
- thermal stress at - 180°C : 254.0 MPa (36.9 Ksi)
- static hullbending stress : 4.9 MPa (0.7 Ksi)

The other point was more difficult to analyse. It was necessary to calculate the stresses at the weld between the washer and the plate, as shown in Figure 7.

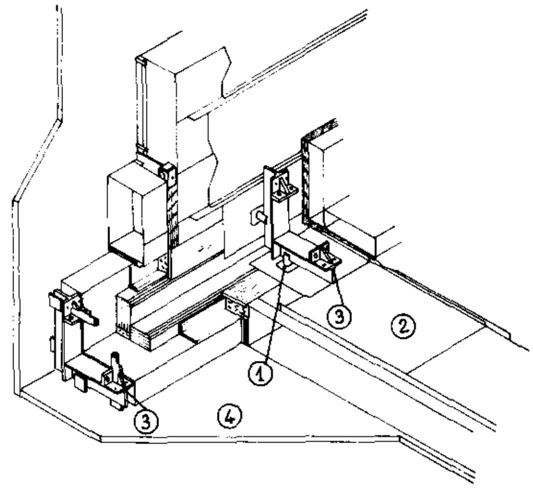

Fig. 7 General view of the bottom corner arrangement
The study detail call washer is the connection between the chair (8) and the secondary membrane (9)

It was then necessary to determine the force which acts on the pipe of the chair. To do this, two models were used.

One was a local representation with an axisymmetric condition of the washer (Figure 9), and the other, a 3 D model of the tank corner (Figure 10).

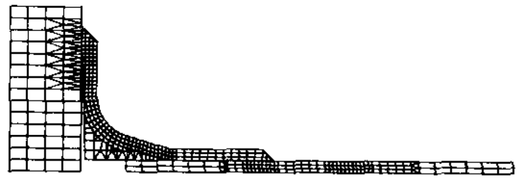

Fig. 9 Axisymmetric model of the pipe, washer and strake-end (one plane view).
This element is shown in Figure 7.

The first model was used to define the rigidity of the connection and the second to define the forces on the pipe and relative displacement.

The 3 D model allows calculations of loads from the hull and loads from tank cool-down.

Also the 2 D model allows calculation of the local thermal stresses and stresses in the weld and at its edges.

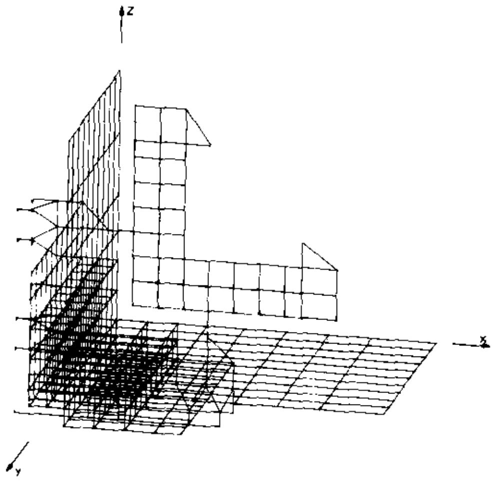

Fig. 10 General view of the 3D finite element model used to analyse the loads on the washer.
This model represents a chair and membrane strake which are defined in Figure 7

The results can be expressed with respect to the forces acting on the pipe, or to dimension problems to a fictious stress or force on the pipe divided by 480 mm^2.

The results of the calculations were as follows :

- wave stress at 10^{-8} probability: 16.8 MPa (2.4 Ksi)
- global thermal stress : 33.7 MPa (4.9 Ksi)
- local thermal stress : 38.3 MPa (5.5 Ksi)
- static hull bending stress : 4.5 MPa (0.6 Ksi).

4 S-N CURVES - DEVELOPMENTS

The problem faced was to determine S-N curves for the two parts under study but with a limited number of laboratory tests.

The Gaz Transport Engineering files provided an important number of results of fatigue tests under constant sine stresses of lapped weld joints with R = 0.1.

The results were given in mean stress in the weld.

From this data, it was possible to develop a S-N curve as shown in Figure 3. The formula obtained by the last mean square method is given by :

$$\sigma^{4.69} N = 6.69\ 10^{14}\ ,\text{for } R = 0.1 \qquad (10)$$

Since the studied connections were lapped joints, it was assumed that the slope of all S-N curves would remain the same.

In the case of the corner piece, the lapped joint was supported, and local bending was not possible as it was for the lapped joints used to establish Figure 3.

A series of tests were conducted on symmetrically supported lapped joints and thin invar plate/thick stainless-steel plate joints.

Two levels of stress ranges were selected, near 10,000 cycles and near 500,000 cycles.

Tests were performed and new mean S-N curves were defined for invar/invar joints :

$$\sigma^{4.69} N = 1.86\ 10^{16} \text{for } R = 0.1 \qquad (11)$$

and for invar/stainless steel joints :

$$\sigma^{4.69} N = 2.14\ 10^{17} \text{ for } R = 0.1 \qquad (12)$$

Also tests were performed at liquid nitrogen temperature and an increase in resistance was found. A correction formula was set up. The S-N curve at a temperature ΔT below ambient temperature which is assumed equal to 20°C, is obtained by multiplication of the log of the constant by the following factor :

$$1. + 0.22 \frac{\Delta T}{216} \qquad (13)$$

The factor is valid between room temperature and - 196°C.

Other tests were performed to study the influence of R. From crack propagation theory, it is assumed that R is limited between 0 and 0.8 to calculate the stress correcting factor.

Therefore tests on lapped joints were performed with R = 0.4 and R = 0.8, at ambient temperature and at liquid nitrogen temperature.

The stress range correcting factor as defined by equation (3) can be expressed by :

$$K = (1 - 1.1\ R)^{-1/3} \qquad (14)$$

Now, for the secondary membrane chair connection, a series of tests were performed for investigation of the stress distribution and stress concentration. The tests were done on exact representation of detail, and the stress range was defined by the fictitious stress, i.e., the force on the pipe divided by the sample plate section, 480 mm^2 .

In order to save time, Locati tests were performed.

For Locati tests, samples are submitted to a series of cyclic loads, increased step by step until failure.

Each step is limited to a maximum number of cycles, 100,000 for the concerned tests. The interpretation has been done by means of the Miner sum and expressed in terms of log C of the S-N curve. An example of results is given in Figure 11.

RESULTS OF TESTS

BB WASHER

STEP NUMBER	1	2	3	4	5	6	7	8
STEP LEVEL (MPa)	36.6	44.7	51.9	61.2	69.4	73.5	77.6	81.7

SAMPLE	NUMBER OF FULL STEPS	LAST STEP NUMBER	CYCLES FOR LAST STEP	C 10^{13}	LOG C
CNA 3	5	6	50 000	11.53	14.06
4	7	8	58 000	27.05	14.43
6	5	6	50 000	11.53	14.06
7	6	7	83 000	20.42	14.31
CNB 1	4	5	70 000	7.40	13.87
2	5	6	72 000	12.77	14.11
5	4	5	39 000	6.06	13.78
GT 1	5	6	35 000	10.68	14.03
2	6	7	65 000	19.10	14.28
3	7	8	10 000	22.51	14.35
4	4	5	33 000	5.80	13.76
5	4	5	70 000	7.40	13.87

Note : The number of cycles for a full step is 100,000.

Fig. 11 Table of experimental results of fatigue tests performed on washer connection models

The resulting S-N curve is given by:

$$\sigma^{4.69}\ N = 1.17\ \ 10^{14} \text{ for } R = 0.1 \quad (15)$$

The above formulae allow the definition of the mean S-N curve. But this curve cannot be used for design purposes since failure occur for stress ranges of lower value. For each series of tests, with the mean value of log C, the standard deviation s has been calculated.

Therefore one must decide on the practical admissible level of failure,i. e., the distance the design S-N curve will remain below the mean S-N curve. This distance is normally expressed in terms of number of s. This number in terms of statistics, corresponds to a certain level of probability of non-failure occurance.

In statistical theories, for the normal law distribution, when the number of samples is very large, a probability of 98 % of non-failure is assumed as representative of a very low level of risk of failure {14} .

This probability leads to a distance between design S-N curve and mean S-N curve of 2 s.

A confirmation of this assumption can be also found in various studies {12,13}. In case of cumulative damage, studies {12, 13} and regulations {15} show that with the S-N curve at 2 s below the mean S-N curve a Miner sum equal to 1 is an acceptable standard to assure that there is minimal risk of failure.

But, the tests are performed on a limited number of samples.

Therefore the distance calculation between the design curve and the mean curve can be improved by statistical laws. It is well known that the mean value follows a Student Law and the standard deviation a X^2 Law. Taking into account a 95 % confidence limit on the estimation and the 98 % probability of failure, the distance is expressed by {14} :

$$d = s\left[\frac{t(.95,n-1)}{\sqrt{n}} + \phi(.98)\sqrt{\frac{n-1}{X^2_2(.95,n-1)}}\right] \quad (16)$$

where :

- t student law
- ϕ normal law
- X^2_2 X^2 law
- n number of samples

The number of tested samples was about 30, therefore, the value of d becomes :

$$d \simeq 3.0\ s$$

The last point to be solved to make the cumulative fatigue analysis possible is the determination of the fatigue limit.

As stated before,the experimental limit is not valid. A precise definition of this limit is consequently not possible. Studies are in progress around the world, and this problem always seems to require intensive research.

Nevertheless a proven method exists by calculating the stress range at $N = 2 \times 10^7$ cycles. But a check can be made by crack propagation analysis to prevent too low and non-realistic levels.

For the material, the threshold coefficient K_{th} can be found from the literature {6} .

It is then possible to calculate the corresponding crack length for a given stress range :

$$K_{th} = \gamma \sigma_1 \sqrt{\pi a} \qquad (17)$$

The crack propagation theory shows that the coefficient γ for the studied disposition can be taken equal to 1.0 {16} .

It is assumed that 80 % of the thickness represents a limit to allow a non propagating crack, it is possible to calculate the corresponding stress limit σ_1 . This level will be considered as a fatigue limit, when greater than the value obtained at 2 x 10^7 cycles on the S-N curve. This stress corresponds to the mean stress in the plate at the edge of the weld.

All of the elements were now in hand to calculate the Miner sum.

A small computer programme was written.

Calculations were performed with the following operational conditions :

- number of tank surveys, two per year.
  In 20 years, n = 40 (full thermal)
- number of voyages, cycles between loaded (partial thermal) and ballast conditions n = 380 (from I.M.C.O.)
- wave stress as given by Figure 4
- design S-N curve at 3 s below the mean experimental curve.

For the corner connections, the found values are as follows, the stresses are given as mean stresses in the weld :

- wave max. stress :
  48.1 MPa (7.0 Ksi)
- thermal stress :
  293.0 MPa (42.5 Ksi)
- total static stress :
  299.0 MPa (43.4 Ksi)
- Miner wave sum :
  .002
- Miner full thermal sum :
  .002
- Miner partial thermal sum :
  .008
- Total Miner sum :
  .012.

For the pipe membrane connection the results are :

- Miner wave sum :
  .00006
- Miner full thermal :
  .0007
- Miner partial thermal :
  .0017
- Total Miner sum :
  .0025

5 CONCLUSIONS

The application of the fatigue analysis method based on S-N curves shows that ship motions and finite element calculations are require, associated with laboratory fatigue tests.

The computer techniques required are nowadays available to all designers or shipyards. Ship motion calculations can be avoided when classification Societies or International Regulations exist. This is especially the case, with LNG carriers, for which the I.M.C.O. code gives formulae as guidance.

Fatigue tests can also be limited if it is possible to find enough data on S-N curves for similar material and joints. In particular, the constant "m" of the equation of the S-N curve can be determined so long as it is a characteris tic of the material. The tests are, therefore, only necessary to determine C and the scatter, which are characteris- tics of the joint type, the weld filler material, the welding procedure and the reference stress used for calculations.

The method is general and can be easily used by designers, but it always shows some inaccuracy and needs to be improved through research work, in particular concerning the fatigue limit in random cumulative fatigue calculation and the probability levels for design S-N curve determination.

The method is a interesting tool available for the analysis of the safety of a structure, with respect to cyclic loads, and it seems logical that such highly sophisticated system as the membrane containment system for LNG carriers, would have been developed through use of such approaches.

Also, an interesting alternative could be a method entirely based on crack propagation. Studies in this direction are in progress, both in the Southwest Research Institute and at Bureau veritas.

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APPENDIX

CUMULATIVE FATIGUE DAMAGE

When a welded joint is exposed to a cyclic load, cyclic stresses and strains are induced in the weld.

Experiments show that failure can be observed for stress levels below the material yield strength, if enough cycles are imposed.

The number of cycles which are necessary to failure is a function of the stress range level.

Therefore, the resistance to cyclic loads, i.e., fatigue resistance, is defined by the relationship between stress range and number of cycles at failure.

This relationship is noted as the S-N curve and for steel materials it can be expressed by the following formula :

$$\sigma^m N = C \quad \text{for } R \qquad (1)$$

where :

- σ stress range
- N number of cycles
- m, c constants, material and joint characteristics
- R ratio between minimum to maximum stress

When a joint is exposed to a series of M loads of different stress ranges σ_i with n_i cycles, the cumulative fatigue damage can be expressed by the Miner sum {1, 2} as follows :

$$\Sigma_1^M \frac{n_i}{N_i} \qquad (2)$$

where :

- n_i number of cycles at stress level σ_i
- N_i N value calculated by the equation (1)

When at step load (i) the value R_i of R is different from the R value of the S-N curve, to calculate N_i it is necessary to correct the stress range σ_i. The stress range must be multiplied by a factor K_i which is a function of R_i.

Therefore N_i is given by :

$$N_i = \frac{C}{(K_i \sigma_i)^m} \qquad (3)$$

When a design is verified it is assumed that the scantling is safe if for an adequate S-N curve the Miner sum remains below an associated value.

As an example, the mean S-N curve is associated with a maximum Miner sum equal to 0.3 {1}, or a S-N curve at 2 standard deviation below the mean curve is associated with a maximum Miner sum of 1.0 {2}.

WAVE STRESS DAMAGE

The stresses induced by wave bending moment are random and are defined by their long term distribution (Figure 6 of the Paper). This curve gives the number of cycles which exceed a given level.

Such a curve can be represented by a step by step function (Figure 4 of the paper). Each step is defined by :

- σ_i stress range
- n_i number of cycles
- R_i value of R
- M total number of steps

The cumulative damage is given by :

$$\Sigma_1^M \frac{n_i}{N_i} = \frac{1}{C} \Sigma_1^M n_i \, (K_i \, \sigma_i)^m \qquad (4)$$

LOCATI TESTS

Locati tests are fatigue tests in which cyclic loads with increasing level step by step, are imposed on the sample, until failure.

Tests are conducted with constant value of R, minimum to maximum stress ratio, and with a fixed maximum number of cycles for each step. In general, a maximum number of 100,000 cycles is used.

When failure is observed after n_N cycles, smaller than the fixed maximum number n, at step M test is stopped. It is then possible to analyse the result.

It is necessary to assume the value m of the equation (1) of the S-N curve, and a value for the Miner sum. The Miner sum value is normally taken equal to 1. The constant C of equation (1) can be, therefore, calculated as follows :

$$C = n \, \Sigma_i^{M-1} \, \sigma_i^m + n_N \sigma_N^m \quad \text{for } R$$

Because the scatter of fatigue results the calculation of C for a given typical weld joint requires various tests, from which the mean value of C and the standard deviation of log C are calculated.

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