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Prediction of Welding Distortion and Panel Buckling of Car Carrier Decks using Database Generated by FEA †

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Abstract

Straightening of welding deformation in shipbuilding is an expensive and time consuming process. Therefore, minimizing welding deformation is required in both design and construction stages. When welding thin plates such as those used in passenger car decks in Car Carriers, welding longitudinal shrinkage, caused by welding tendon force, causes compressive stress in the surrounding plate fields that sometimes causes these plate fields to buckle. Straightening of a buckled plate is difficult and buckling should be avoided whenever possible.

For this purpose, a cross stiffened panel of a car deck of a Car Carrier is considered. A series of thermal elastic plastic finite element analyses is carried out to predict the welding tendon forces of Longitudinals and Transverses when utilizing continuous and intermittent welding with different welding process specifications. Bi-directional residual compressive stresses are evaluated and buckling is checked.

Results indicate that when the geometry of the welded cross-section is similar, welding tendon force depends almost only on the average heat input per unit length of the weld line, no matter whether welding is continuous or intermittent. The effectiveness of parallel and zigzag intermittent welding in reducing welding residual stress and preventing buckling is quantified. A tendon force database may be created such that designers may easily check buckling and select welding specifications

KEY WORDS: (Welding Deformation) (Welding Procedure) (Tendon Force) (Buckling) (Deflection) (Thermal Elastic Plastic Analysis) (Plate Width Effects) (Edge Effect)

1. Introduction

During ship hull construction, welding is extensively used to join stiffeners to plates, build subassemblies, assemblies and blocks, and then to join these blocks to assemble the ship hull. This welding causes unavoidable deformation. One of welding deformation modes is the out-of-plane deformation of plate fields. When this deformation exceeds permissible limits, it is reduced using some straightening process, mainly spot and line heating. Straightening is expensive and time consuming because it is done manually. To reduce cost and time spent in straitening, it is desirable to minimize welding deformation, not only in the production stage but also in the design stage. If welding deformation under a given design and a production procedure is predicted, measures to reduce deformation can be taken. Therefore prediction of welding deformation with appropriate accuracy is a key to achieving such cost and time reduction.

From straitening point of view, one of the most difficult welding deformation problems is buckling. Buckling may occur under the compressive residual stresses arising from the longitudinal shrinkage caused by welding tendon force. Thin plate stiffened panels, such as car decks of a Car Carrier, often have not only general welding deformation but also such buckling deformation. Straightening of a buckled plate is difficult and time consuming. To avoid this problem, it is necessary to accurately predict welding residual stress, check plate buckling and avoid it whenever possible. In this study, the influences of the geometry and welding procedures on the tendon force are studied through a series of computations using a three dimensional thermal elastic plastic finite element analysis tool that utilizes the Iterative Substructure Method[1)].

2. Validity of computation

To check the validity of numerical analysis in this

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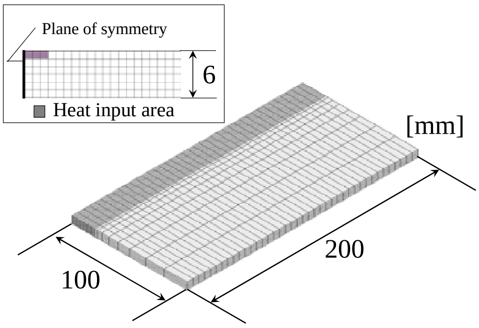

Fig.1 Analysis model to examine validity of simulation.

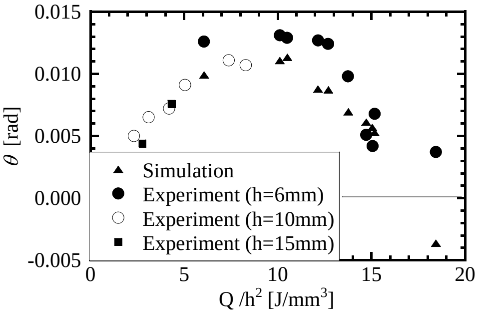

Fig.2 Comparison of angular distortion between experiment and simulation.

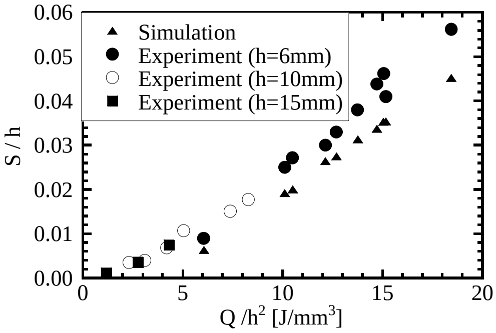

Fig.3 Comparison of transverse shrinkage between experiment and simulation.

study, computed results of transverse shrinkage and angler distortion are compared with experimental results of bead on plate welding conducted by Sato and Terasaki[2)]. The plate is 200 x 200 x 6 mm. Due to symmetry, only one half of the plate is analyzed. Finite element model is show in **Fig.1**. Plate size, material properties and heat input are the same as in the experiments.

Figures 2 and **3** show comparisons between computed and experimental results. From these figures, it may be seen that the differences between computed results and experimental results are small and that the numerical analysis has enough validity.

3. Extent of Finite Element models and boundary conditions

Welding induced buckling of stiffened panels is

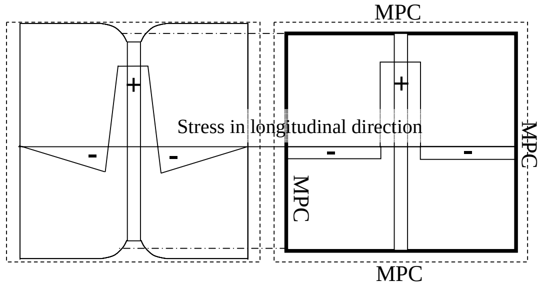

Fig.4 Schematic comparison of deformed plate in the cases without and with multi point constraint (MPC).

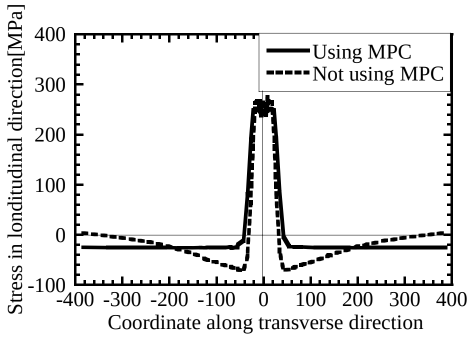

Fig.5 Comparison of longitudinal stress distributions in the cases with and without multi point constraint (MPC).

caused by residual stresses produced during welding. When stiffeners are welded to a plate, the plate shrinks in both the longitudinal and the transverse directions. In this study, longitudinal shrinkage, that is the Tendon Force is considered to be the main cause of buckling. Finite element simulation of welding of a large stiffened plate with many stiffeners is difficult due to the large computation resources that it needs. Therefore, it is preferable to consider appropriate parts of the stiffened plate with appropriate boundary conditions. Extents (sizes) of models used in this study are examined later in this paper. In this study, a “Multi Point Constraint (MPC)” is used for in-plane boundary condition at edges. By this constraint, the edges is kept straight but free to move in the plane of the plate parallel to its original position as shown in **Fig.4**. This well reflects the restraint given by the rest of the panel surrounding the considered part. A comparison of the longitudinal stress obtained with free edges and with this MPC is shown in **Fig.5**. Besides being a correct representation of constraint, this boundary condition results in some advantages. It becomes easier to predict the compressive residual stress of a large panel using a smaller part of the panel, even if the welding is not continuous, such as zigzag or parallel intermittent welding. In this case average value of welding compressive residual stress can be directly predicted from the shrinkage of the plate between the edges of the small model.

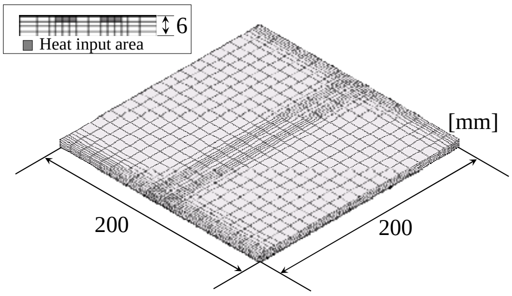

Fig.6 Plate model to predict tendon force.

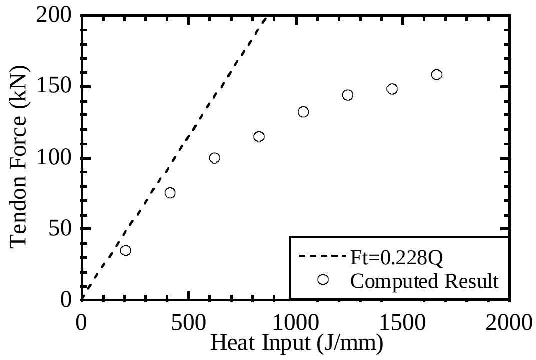

Fig.7 Influence of Heat Input on Tendon Force.

4. Prediction of the Tendon Force

4.1 Influence of Heat Input on Tendon Force

To clarify the influence of heat input on tendon force, a series of computations are carried out using the finite element model shown in **Fig.6**. Standard welding voltage, current, and speed are taken as the average of those actually used in shipyards for this plate thickness, that is 225 A, 24 V, and 6.5 mm/sec, respectively. Heat efficiency is taken as 0.75 leading to standard net heat input of 623 J/mm. To examine the influence of the heat input, the welding power is changed around the standard value while keeping welding speed constant at 6.5mm/sec. The (solid) dotted line in **Fig.7** is the commonly used tendon force formula

Tendon force : $F_t = 224 Q_{net}$ (1)

proposed by Terasaki and others[3]. In this equation, the heat input Q_{net} represents the net heat input. Results of numerical analysis are also plotted in the figure. As may be seen from the figure, as heat input increases, the computed tendon force becomes significantly smaller than that predicted by the formula. This is because as heat input, and consequently temperature increase, heat transfer from plate surface increases and heat is not effectively used.

Therefore, in this study it is decided not to use the formula of **Eq.(1)** and detailed thermo elastic plastic analyses are used to predict the tendon force.

4.2 Selection of Model Size

As pointed out before only appropriate parts of a stiffened plate is to be considered with appropriate boundary conditions. To find out the necessary model size with respect to the tendon force, a series of computations is carried out with three different plate sizes, a complete plate panel of a passenger car deck of a car carrier (3280 x 820 x 6 mm), a plate with a small length (500 x 820 x 6 mm) and a plate with small length and width (500 x 500 x 6 mm). FE models are shown in **Figs.8**, **9** and **10**. The same multi point constraint boundary conditions are used in all cases. Welding heat input is varied as in 3 above.

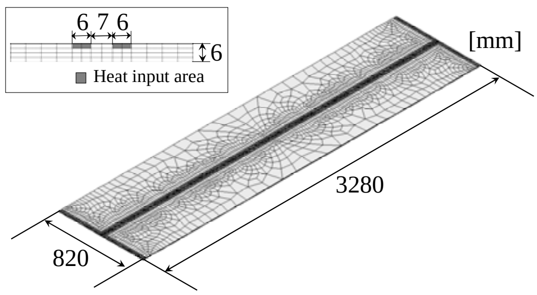

Fig.8 Full scale plate model.

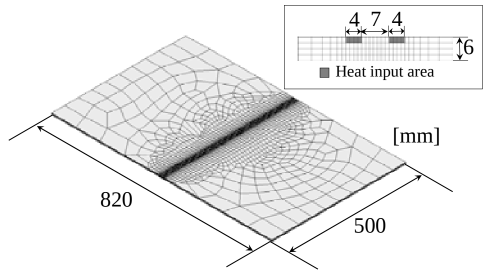

Fig.9 Plate model with small length.

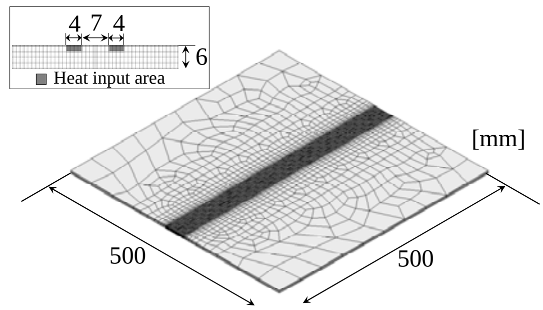

Fig.10 Plate model with small length and width.

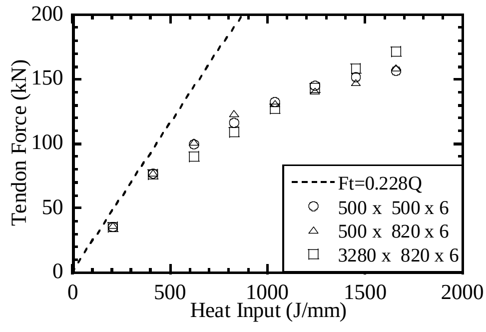

Fig.11 Influence of size of plate on tendon force.

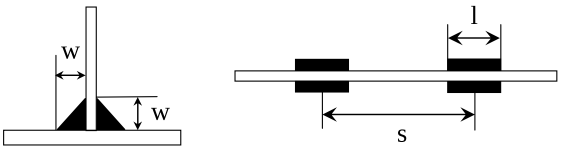

Fig.12 Illustration of intermittent welding.

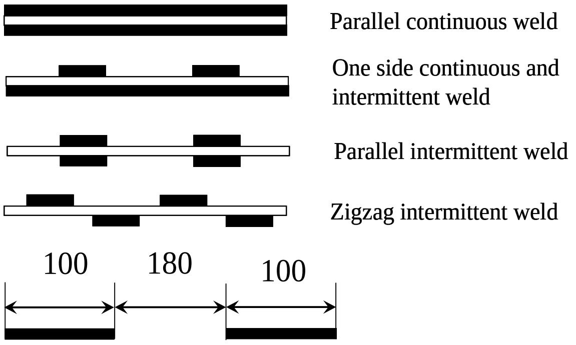

Fig.13 Illustration of commonly used welding procedure.

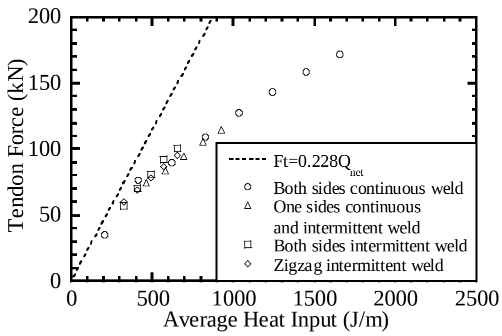

Fig.14 Influence of welding procedure on tendon force.

Results of analysis are summarized in **Fig.11**. It is clearly seen from the figure that the influence of the plate size on the tendon force is not significant. Therefore, when using appropriate boundary conditions, it is possible to use a model of a small part of the plate (500 x 500 x 6 mm) to evaluate the tendon force instead of using a model of the whole plate.

4.3 Influence of Welding Procedure

In actual welding of many ship structural components, intermittent welding is employed instead of continuous welding to reduce welding distortion and cost. For example, The American Bureau of Shipping allows intermittent welding as follows when welding stiffeners to plating in certain locations, including passenger car decks in Car Carriers, see **Fig.12**.

$$s - l \le 32 t_{pl} \tag{2}$$

$$w = t_{pl}\, C \frac{s}{l} + 2.0\left(1.25 - \frac{l}{s}\right) \tag{3}$$

Where,

w : Weld leg length;

l : Actual length of weld fillet, clear of crater;

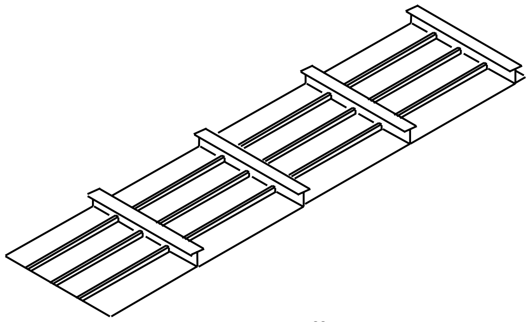

Fig.15 Continuous stiffened plating.

s : Distance between successive weld fillets from center to center of welded parts;

t_{pl} : Thickness of the thinner of the two members being joined;

$C = 0.12$: Weld factor of longitudinal stiffener;

$C = 0.15$: Weld factor of transverse stiffener.

In this study, leg length is taken equal to 4 mm as in actual shipyard practice. The welded length l and unwelded length ($s - l$) are taken as 100 mm and 180 mm respectively satisfying **Eqs.(2)** and **(3)**. Three kinds of intermittent welding are considered as shown in **Fig.13**, and compared with continuous welding.

To clarify the influence of welding configuration on the tendon force, a series of computations is carried out using thermal elastic plastic FEM.

Computed results are summarized in **Fig.14**. In this figure, the tendon force is plotted against the average heat input per unit total weld length (total weld length includes unwelded lengths ($s - l$)). From this figure, it may be seen that the tendon force is almost the same when the average heat input per unit total weld length is the same, whether welding is intermittent or continuous. This means that, for the same joint cross section configuration, the tendon force is dependent only on the average heat input per unit length. Therefore, where weld leg length can not be reduced beyond practical limits or design rule requirements, intermittent welding would be effective in reducing the heat input and consequently the tendon force and residual stress.

4.4 Influence of Stiffeners on Tendon Force

Continuous stiffened plating (see **Fig.15**) of a car deck of a car carrier is considered. The plate thickness is 6 mm. Transverses are spaced at 3280 mm and longitudinals are spaced at 820 mm. To predict the tendon forces produced by welding of the transverses and longitudinals of the actual stiffened panel, a series of computations are carried out using the models shown in **Figs.16** and **17**. Model edge length in both the longitudinal and transverse directions is taken as 500 mm, since the tendon force does not appreciably change

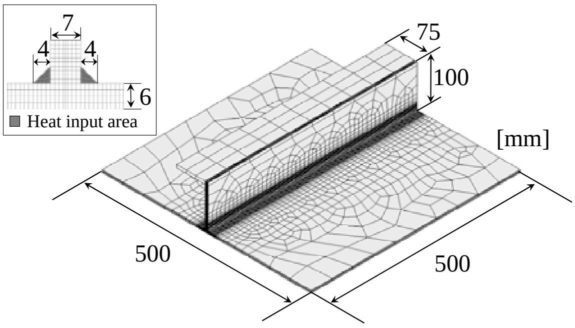

Fig.16 Analysis model of actual longitudinal stiffened model.

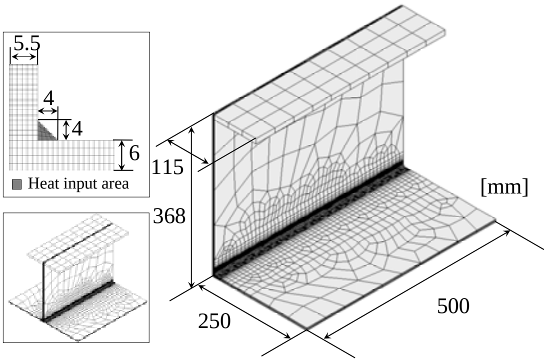

Fig.17 Analysis model of actual transverse stiffened model.

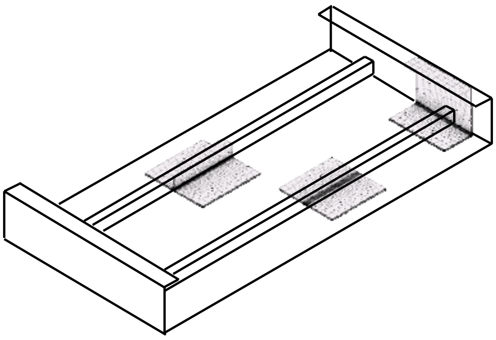

Fig.18 Position of each model in car deck.

with plate size. Longitudinal stiffeners are L-sections with web and flange thickness of 7 mm and modeled as in Fig.16. Transverse stiffeners are T-sections with web and flange thicknesses of 11 mm and 12 mm respectively and modeled as in Fig.17. One half of the model with a transverse stiffener is used in the computations because of symmetry.

Computed tendon force F_t is plotted in **Fig.19** with respect to the heat input Q_{net}. Tendon force of bead on plate welding is also plotted for reference. In case of fillet welding, 8 cases with different heat input Q_{net} are analyzed. The heat input per unit weld length that corresponds to the welding condition employed in the shipyard is taken as a standard value. Average heat input is varied from 12.5 % (corresponding to low heat intermittent welding on one side of the stiffener) to 133 % (corresponding to high heat continuous welding

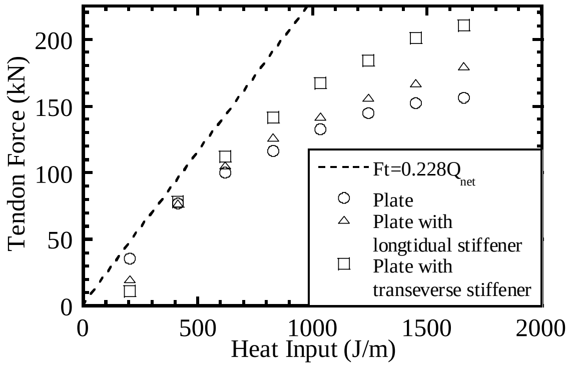

Fig.19 Residual Stress after each Stiffener's Welding.

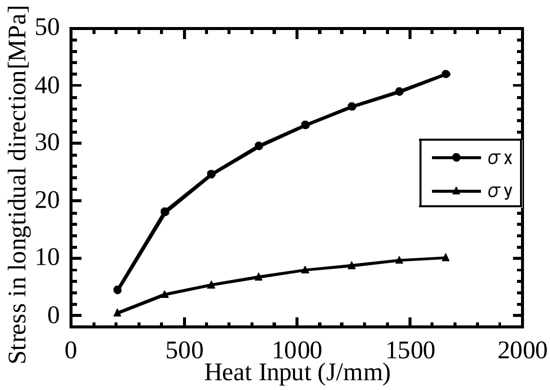

Fig.20 Welding residual compressive stress in the longitudinal and transverse directions.

on both sides) of the standard value. It may be seen that the Tendon force of the stiffened models is larger than that of the plate model. This is because stiffened models have stiffeners and weld fillets. Therefore they have larger heat capacity than that of the plate model. This lowers welding maximum temperature. Also the ration of heat conduction to the rest of the model is increased because of the existence of the stiffener. Therefore the heat transfer to the air is relatively smaller and more heat input is effectively used into creating the tendon force.

5. Buckling Check

5.1 Residual Stress calculated from Tendon Force

Average residual biaxial compressive stress is calculated from the tendon forces Ft of the longitudinally and transversely stiffened models using **Eq.(4)** and plotted in **Fig.20**. σ_x is the average longitude compressive stress produced when welding the longitudinal and σ_y is the average transverse compressive stress produced when welding the transverses.

$$\sigma = \frac{F_t}{S} + \frac{(F_t \times h) \times h}{I} \tag{4}$$

Where,

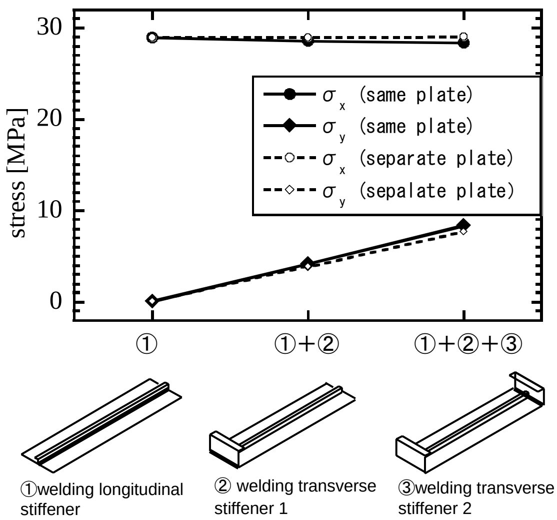

Fig.21 Influence of some welding in one plate.

h : Distance between the application point of F_t and cross section neutral axis;
S : Cross sectional area of plate and stiffener;
I : Moment of inertia of plate and stiffener;

5.2 Residual Stress after each Stiffener's Welding

The residual stresses plotted in Fig.20 are calculated by welding a longitudinal and a transverse each on a separate stress-free plate. In actual welding of stiffeners to plating, longitudinals are welded first, and then transverses are welded. The difference of residual stress produced after welding of longitudinals then transverses on the same plate, and the sum of the residual stress produced by welding of longitudinals and transverses on separate plates is checked. This is carried out using a plate panel measuring 3280 x 820 x 6 (mm) and standard net heat input of 623 (J/mm).

Figure 21 shows that the difference of residual stress between the two cases is very small. Therefore, residual stress of a stiffened plate produced by welding of longitudinals then transverses can be predicted as the sum of those residual stresses produced by welding the longitudinals and the transverses on separate plates.

5.3 Buckling Check

Figure 20 may be used to predict the compressive stresses σ_x , σ_y resulting from welding of the longitudinal and transverse stiffeners shown in Figs.16 and 17 respectively, to a 6 mm thick plate. From these compressive stresses, plate buckling may be checked using the classical equation of buckling of simply supported rectangular plates compressed in two perpendicular directions as given by the following equations[8].

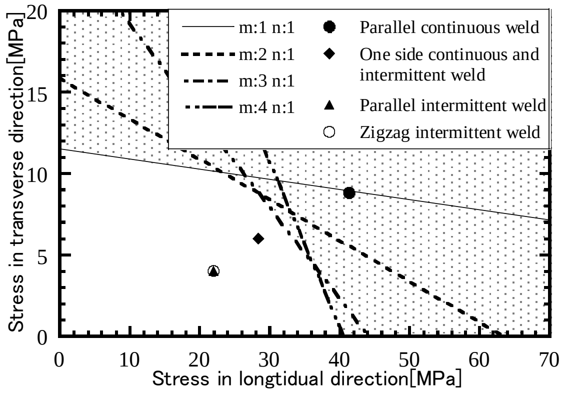

Fig.22 Buckling check using predicted residual stress.

$$\sigma_x m^2 + \sigma_y n^2 \frac{a^2}{b^2} = \sigma_e \left(m^2 + n^2 \frac{a^2}{b^2} \right)^2 \tag{5}$$

$$\frac{\pi^2 D}{a^2 h} = \sigma_e \tag{6}$$

$$D = \frac{E h^3}{12\left(1 - \nu^2\right)} \tag{7}$$

Where,

a = Plate length;
b = Plate breadth;
σ_x = Longitudinal stress;
σ_y = Transverse stress;
m = Number of half buckling waves in the longitudinal direction;
n = Number of half buckling waves in the transverse direction;
E = Young's Modulus;
ν = Poisson's ratio;

Compressive stresses obtained from this figure are used to examine the buckling of a car deck plate panel 3280 x 820 x 6 mm. Buckling stress interaction curves of modes (m, n) and the compressive stress produced by different types of fillet welds of the longitudinal and transverse stiffeners are summarized in **Fig.22**. From this figure it may be seen that "both sides continuous" weld causes buckling of the plate. "One side continuous and intermittent weld" is very close to buckling stress. The plate may buckle when further force is applied, as for example, during block assembly. On the other hand, intermittent welding reduces the residual compressive stresses and hence buckling of the plate can be prevented.

6. Conclusion

A series of thermal elastic plastic finite element analyses is carried out to predict welding residual stresses and plate buckling of passenger car decks of a Car Carrier. Results of analysis suggest that;

(1) The effect of plate size on welding tendon force is very small, therefore, the tendon force of a large welded structure can be predicted by using small models;
(2) When the geometry of the cross-section of a welded joint does not change, the tendon force depends only on the average welding heat input no matter whether continuous or intermittent welding is used;
(3) Parallel intermittent welding and zigzag intermittent welding can reduce heat input and welding residual stress, and hence buckling can be prevented.

Acknowledgements

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