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V. B. Nguyen

P. K. C. Wood

M. A. English

M. A. Castellucci

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The design and development of new cold roll formed products by finite element modelling and optimisation

V.B. Nguyen[1], P.K.C. Wood[2], M.A. English[3] and M.A. Castellucci[4]

Abstract

The design and development of new cold roll formed products can incur significant cost and the product may not be optimised for either performance or manufacture. This paper describes a new method to develop an optimum structural design of profile by cold roll forming using a combined approach of finite element analysis and optimisation techniques. To illustrate the concept, the design and development of a new channel beam and a new drain grating subjected to bending are presented. The two case studies, demonstrate how a roll formed profile may be optimised to improved structural performance through use of stiffeners and/or dimples. Improved performance of cold roll formed products is achieved by increasing the strength of the product without increasing the amount of the material used. The results of this paper clearly demonstrate an efficient and effective method and tool set to optimise design for performance and manufacture of cold roll formed products.

[1]Lecturer, College of Engineering and Technology, University of Derby, Markeaton Street, Derby, DE22 3AW, UK.
[2]Reader, Institute for Innovation in Sustainable Engineering, University of Derby, Quaker Way, Derby DE1 3EE, UK.
[3]Design and Development Manager, Hadley Industries plc, Smethwick, West Midlands, UK.
[4]Group Technical Director, Hadley Industries plc, Smethwick, West Midlands, UK.

Introduction

In the cold roll forming industry, there is a critical requirement that is to reduce the initial strip to a minimum while maintaining the structural performance of the roll formed products, thus minimizing the major financial outlay in the process which is the material cost. The development of various alternative cold rolled formed profiles which improve the structural performance of the section by including additional bends such as ‘intermediate stiffeners’ or ribs or dimples (Rhodes and Zaras 1988, Nguyen et al. 2011), as shown in Figure 1, has been a solution for these conflicting requirements. These stiffeners subdivide the plate elements into smaller sub-elements and hence can considerably increase the local buckling strength of cold-formed sections subjected to compressive stresses; it is because of smaller width-to-thickness ratio of the sub-elements. The zed section with longitudinal stiffeners in the web, introduced during cold rolled forming, was designed and developed at the University of Strathclyde by Rhodes and Zaras (1988) in conjunction with Hadley Industries plc, with the aim of improving a zed type section. The development suggested that when the stiffeners were placed about one fifth of the web width from each flange, the problem of local buckling in the web was eliminated. The channel section with longitudinal stiffeners in the web was developed at Hadley Industries plc later in an attempt to incorporate the innovative web stiffener configuration used in the new zed, into a channel shape (Castellucci et al. 1997).

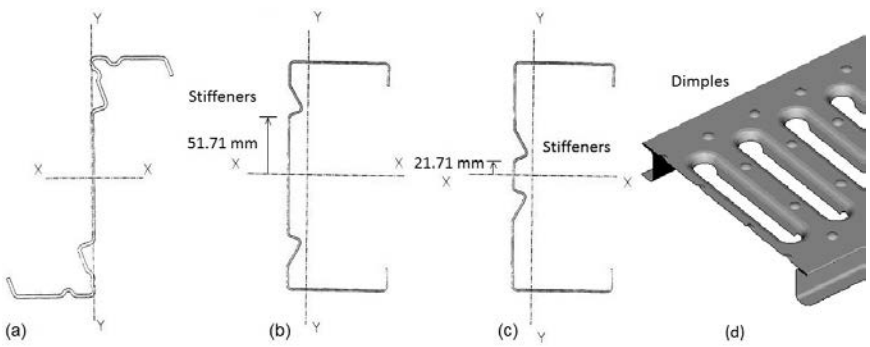


Figure 1 Intermediate stiffeners in (a) zed, (b) and (c) channel, and (d) grating

In recent years, there has been a significant amount of studies on the strength and design of cold-formed sections with web stiffeners (Desmond et al. 1978, Papazian et al. 1994, Schafer and Pekoz 1998, Young and Chen 2008, Zhang and Young 2012). However, there has been limited investigation on optimum design of a section, considering the effects of location and shape of stiffeners on the section subjected to bending.

Owing to the complex and interrelated nonlinear changes in contact, geometry and material properties that occur in the process and product forming, theoretical and design calculations cannot be used to accurately analyse the performance of the products with additional bends or dimples. These, however, can be solved by using a finite element (FE) modelling approach which is capable of simulating complicated processes and products (Nguyen et al. 2013, 2014). This allows optimisation of the process and subsequent products in order to improve the product structural performance or to reduce the product material.

In this paper, finite element simulations and optimisation techniques were presented as tools for new process and product developments and illustrated through case studies of optimisation of cold roll formed products. The design and finite element simulations, using two practical case studies, were carried out in three stages: (1) Developing new product geometries from a proposed / existing ones by varying their geometric parameters, against the target performance of the product, using parametric modelling technique via the finite element package PATRAN (MSC Software), (2) Planning the design of experiments (DOE) using a response surface model, running multiple simulations, recording the performance of the system at each run and determining geometric values that give the target performance: a maximum strength to weight ratio - these procedures are done using ADAMS/INSIGHT and MARC solver (MSC Software), and (3) Simulating the mechanical tests of new products and comparing with recently conducted product test results for validation. The two case studies of this FE and optimisation techniques included the design and development of a new channel and a steel grating.

Development of a new channel

Finite element analysis and optimisation procedures

The original shape of the channel had no stiffeners and the new channel had two stiffeners positioned at an equal distance to the web centre as shown in Figures 1. The section has a web depth of 170 mm, a flange width of 63 mm and a thickness of 1.60 mm. The steel material has a yield stress of 519 N/mm^2 and a tensile strength of 550 N/mm^2. The position of the two stiffeners influences the channel's strength in a 4-point beam bending test. Braces at close gap (200-300 mm) were used to ensure local buckling occurred in the beams. In the simulation, the position of the stiffeners was changed from a minimum value of 21.71 mm (Figure 1(c)) to a maximum value of 51.71 mm (Figure 1(b)) with 10 different values in between.

Figure 2 illustrates the model setup for FE analysis that includes boundary conditions and loading configuration. By taking advantage of symmetry, only a half of the test system was modelled. The simulations were carried out on beam specimens, simply supported at both ends. The beam mesh was defined as a function of the section width and element size (of 4 mm) so that when the section shape changed due to changing the position of the web stiffeners, the mesh size and number of the beam were retained. When the stiffeners were placed at a maximum value of 51.71 mm, the channel beam was modelled using 83,220 elements and 84,073 nodes; they are four-node, thin-shell elements with global displacements and rotations as degrees of freedom (element type 139). The braces were modelled as rigid links connections. Load was applied on the two central cleats at their centroids using the displacement-controlled method while the two end supports were fully fixed in vertical direction at their centroids. Each loading point was at a reference node that connects to a set of tied nodes (at the beam web where the cleat connected to the beam). The link between the reference node and the tied nodes was based on a rigid link connection, only unrestrained in loading direction. The displacement was increased in successive increments until the beams failed. A full Newton-Raphson method was used for the iterative procedure and an implicit, static analysis was employed. Simulations of the beams in bending test were undertaken in two steps. In the first step, a linear elastic buckling analysis was performed on the perfect beam to obtain its buckling mode shapes (eigenvalues). In the second step, a nonlinear post-buckling analysis was carried out to predict the beam post-buckling behaviour and ultimate load capacity. The shape of initial geometric imperfections and magnitude as suggested in Nguyen et al. (2013) was taken to generate the initial imperfections for FE analysis; it deemed to be similar to the mode observed in tests (for example, FE failed modes were compared with experimental ones, as shown in Figure 4).

In this analysis, the input parameter was assigned for the position of the web stiffeners and the output parameter was the buckling strength of the beam.

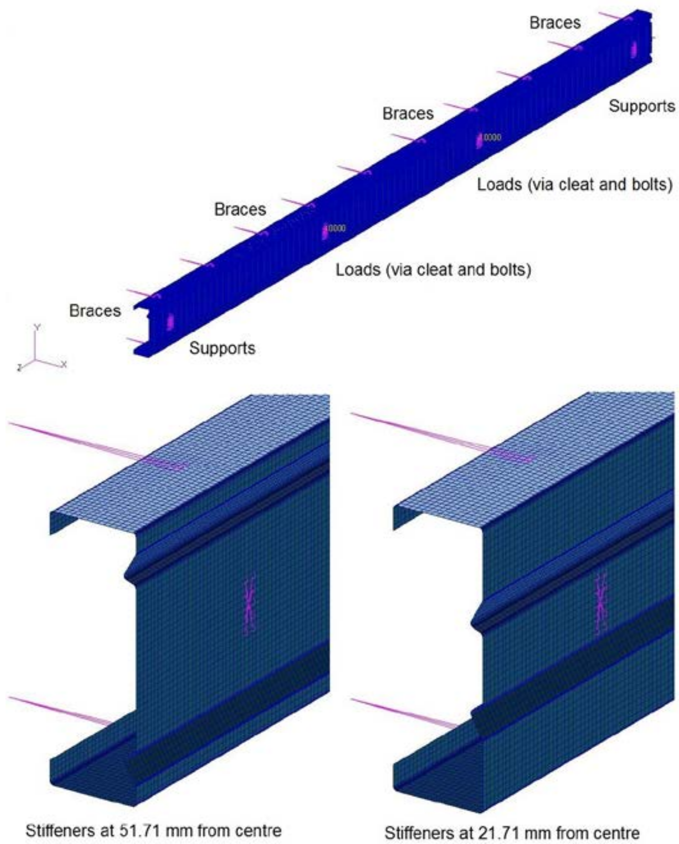


Figure 2 FE model of a 4-point bending test setup including boundary conditions and a closer view of the meshes at two different positions of the web stiffeners

Results and validation

Buckling loads were obtained as the target performance and it was found that buckling load increased linearly with increasing position of the stiffeners from the minimum position to the maximum position, through 10 different positions of the stiffeners. The optimum case was achieved when the two stiffeners positioned at 51.71 mm to the web centre with a maximum buckling load of 45.20 kN (compared with 44.04 kN when the two stiffeners positioned at 21.71 mm). Hence, the channel with two stiffeners positioned at 51.71 mm to the web centre was developed and named UltraBEAM™2.

To validate the FE simulations, FE and experimental load-displacement curves of the 4-point bending test of the UltraBEAM^{TM}2 channels are shown in Figure 3. The FE and experimental results were close in both buckling and ultimate loads, with a maximum difference of less than 5% in buckling load and 6% in ultimate load. The failed mode shape of the channel is shown in Figure 4, in which the experimental shapes were also illustrated for validation. The comparisons show excellent agreements between simulation and test.

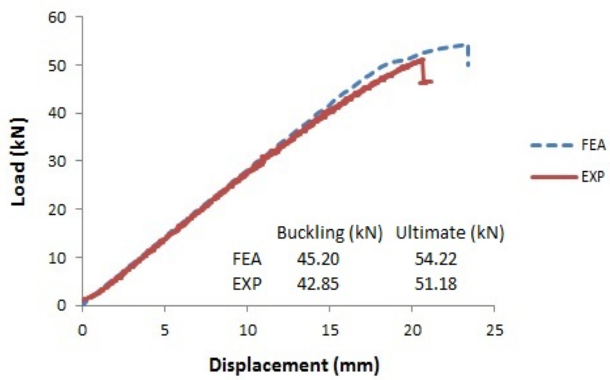


Figure 3 FE and experimental curves of 4-point beam bending test of UltraBEAM^{TM}2 channels

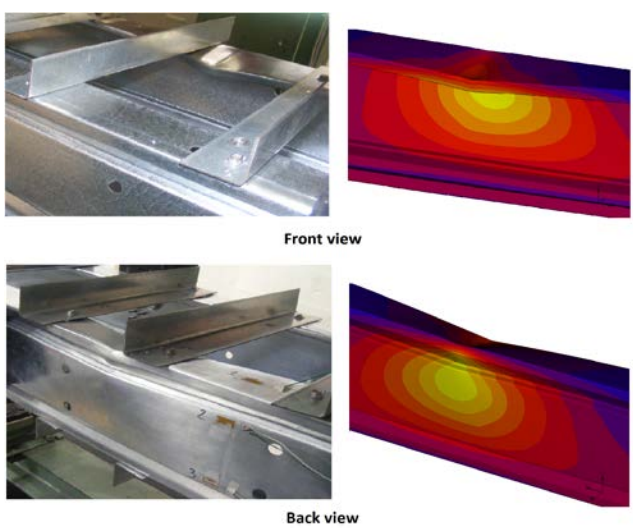


Figure 4 Failed mode shapes of the UltraBEAM^{TM}2 in testing and simulation

Development of a new grating

Finite element analysis and optimisation procedures

In this study, the FE simulation of an existing grating was carried out and the results compared with experimental results to validate the simulation setup. The existing grating had a length of 499.50 mm, a width of 128.30 mm and a thickness of 0.935 mm. Load on the grating was applied through the rigid load plate, similar to the condition in real test setup, as illustrated in Figure 5. The results in Figure 5 show that there is a good agreement between the FE and test results with a maximum difference of 6% for ultimate load (load at a displacement of 10 mm).

The validated FE model was then extended for modelling and developing a new grating product, as shown in the model setup in Figure 6. The performance of the grating was examined by changing parameters including: dimple height (scale: 1 – 2), dimple width (1 – 4 mm), slot length (2 – 20 mm), slot width (1 – 4 mm), plate thickness (3 – 5 mm), plate height (scale: 0.65 – 1), as illustrated in Figure 7. The output target results were strength in terms of maximum stresses and deflections of the grating.

The grating mesh was defined as a function of the section width and element size (minimum of 0.20 mm for the dimple elements, and 1.20 mm for elements outside the dimple) so that when the section shape changed due to changing the model parameters, the mesh size and number of the beam were retained. When the parameters assigned their maximum values, the grating was modelled using 99,976 elements and 113,091 nodes; they are four-node, thin-shell elements with global displacements and rotations as degrees of freedom (element type 139). The rubber pad was used to largely spread the load from the steel block (placed underneath the load cell) to the grating and it was modelled using 28,800 solid elements; they are 3-D eight nodes hexahedron elements. The steel block was modelled as a rigid load plate moving vertically in the negative vertical direction to a predefined displacement; for the purpose of this study, a small displacement of 1 mm was used and all the responses were compared at this applied displacement. The contact between the grating, rubber and plate was modelled as contact surfaces using 3D contact elements. ‘Glued’ contact was used for contact between rubber pad and rigid load plate. ‘Touching’ contact between the grating and the rigid support plate, the grating and rubber pad and of the grating itself were defined. It was assumed that there is frictionless contact between the grating and plate. A full Newton-Raphson method was used for the iterative procedure and an implicit, static analysis was employed. Large strain nonlinear procedure was used to take into account geometric and material nonlinearity.

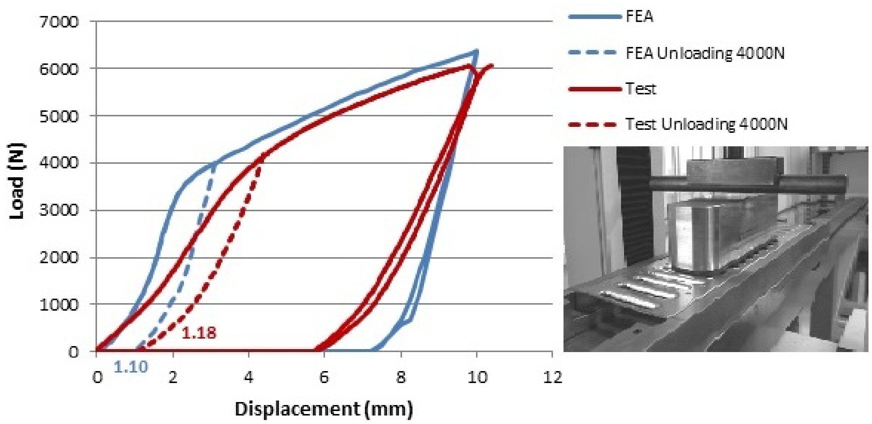


Figure 5 FE and experimental force-displacement curves of the grating. FE model setup of the existing grating similar to test setup (in box)

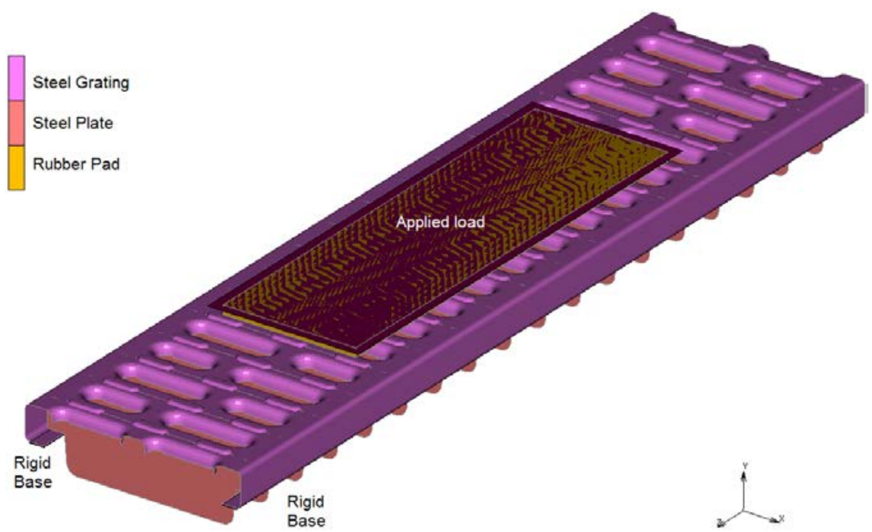


Figure 6 FE model setup of the new grating including load path and boundary conditions

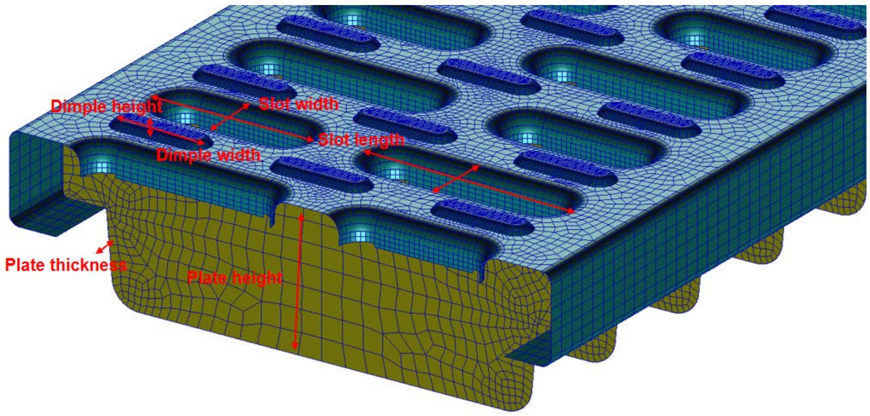
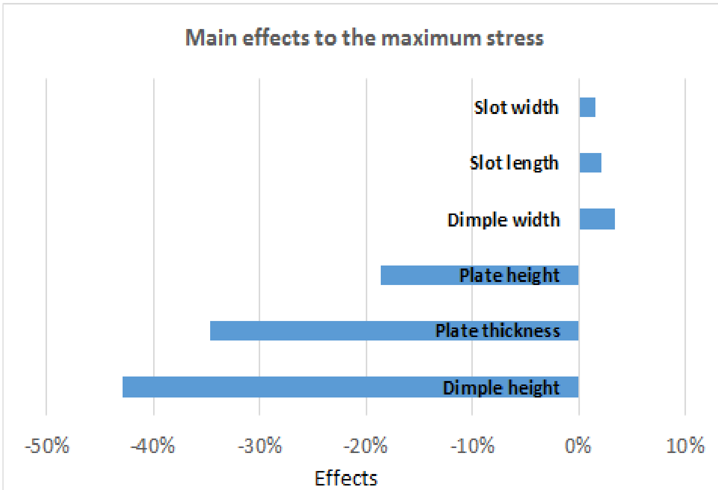


Figure 7 FE mesh of the new grating which shows the studied parameters

The process of varying all the parameters was carried out in the program ADAMS/INSIGHT. In which each parameter was assigned three different values in the range from min to max value. The target responses selected in this study were maximum tress and minimum deflection in the grating. Each response was a response surface function of all parameters, and was treated as an objective. There were a total 729 runs integrating all parameters while the applied loads on the gratings were the same for all the runs.

Results and discussions

It was found that dimple geometries are the most effective parameters to both stress and deflection, as shown in Figure 8. In this figure, 'Positive' Effect % means response increases with larger parameter value, and 'Negative' Effect % means response decreases with larger parameter value. In this study, 'Positive' Effect for the maximum stress response means the stress increases while 'Negative' Effect for the deflection means the deflection decreases, and vice versa. It can be seen that increasing 'dimple height' is the most effective way to increase the grating strength and reduce its deflection, with up to 44% effect; increasing 'plate thickness' is also one of the most effective solution. However, increasing 'plate height' parameter value is not effective for both responses; therefore, the plate height can be reduced to save the grating weight or material.



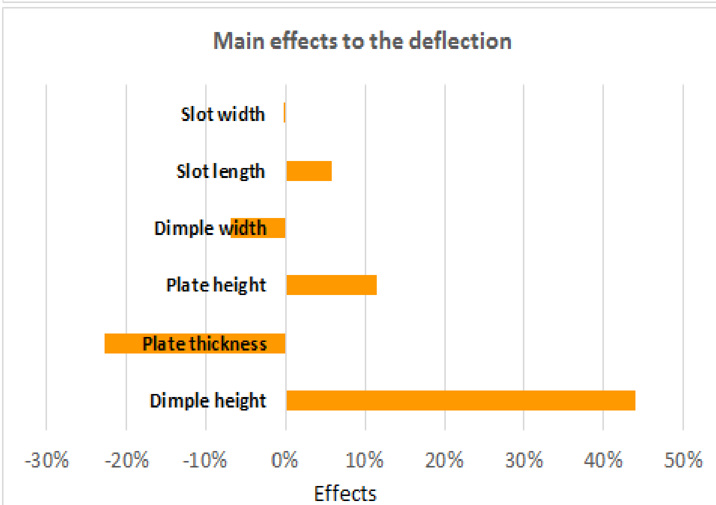


Figure 8 Effects of parameter values to the grating responses: the maximum stress (above) and the deflection (below)

These observations can be also seen in Figure 9 which shows the model of central part of one grating (out of 729 runs) in which maximum stresses developed around the dimples whilst less stresses generated in the plate.

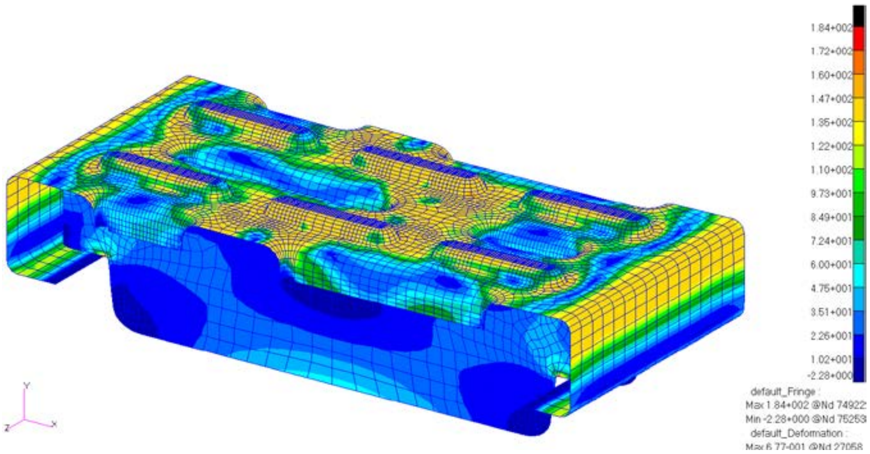


Figure 9 Stresses distribution in the central part of the grating at a predefined displacement

Based on these results, the aim of designing the grating is to find a balance for the dimple height and plate thickness that can give the maximum strength and minimum deflection in the grating. With specified target responses (stress and deflection that satisfied the standard test requirements), a set of parameter values were determined, and an optimum design of the grating was achieved.

Conclusions

This paper has presented the design and development of new cold roll formed products by using a combined approach of the finite element analysis and optimisation techniques to simulate the products' structural responses and obtain the optimum design for the products. Two case studies which included a new channel beam and a new drain grating subjected to bending were presented to illustrate the design concepts. For the channel UltraBEAM^{TM}2, the development suggested that when the longitudinal stiffeners were placed on the web as much close as possible from each flange, the buckling and ultimate strengths of the beam could be maximised comparing with other positions. For the new grating development, it showed that when a set of 'most effective' parameter values were determined, i.e. the dimple height and the plate thickness, and an optimum design of the grating could be achieved. This study demonstrates that the finite element modelling together with optimisation techniques provide powerful practical tools to analyse and obtain optimum design for complex products. The successful simulations could enable the cold roll forming industry to provide novel cold roll formed products or alternative products with stiffeners which are developed from optimum design concepts.

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