Materials Science-Poland, *Vol. 28*, *No. 1*, *2010*

Finite element stress analysis of forging dies to improve their fatigue life

K. DEHGHANI[1*], A. JAFARI[2, 3]

[1]Amirkabir University of Technology, Tehran, Iran

[2]Shahid Bahonar University of Kerman, Kernan, Iran

[3]University of Tehran, School of Mechanical Engineering, Tehran, Iran

The forging dies, and in a particular case a bolt die, were stress analyzed by the finite element method. Two possible modes of die failure, due to hoop and axial stresses, were investigated. The critical zones of highly concentrated stresses have been identified. Several approaches were studied to overcome the tensile stresses that result in a premature failure of a forging die. The results of the finite element simulations show that generating the compressive or negative stresses, as produced by the techniques applied, can completely remove, or at least significantly reduce the detrimental tensile stresses generated during forging. This can readily improve the fatigue life of dies. Numerical stress analysis was performed on critical elements lying in the transition zone of dies. Finally, advanced numerical methods, especially the finite element method, were used to determine the optimum mean stress and the optimum alternative stress as well as to analyze the compressive negative stresses generated by the applied techniques. The ABAQUS software was used for the finite element simulation. The optimum mean stress and the optimum alternative stress at the most critical finite element were determined to be 140–150 and 34–38 MPa, respectively.

Keywords: *die-life improvement; forging; extrusion; pre-stress; fatigue fracture*

1. Introduction

Forging is among the best techniques for the production of tools. However, the major concern is the fatigue life of forging dies, so that not only the selection of an appropriate material is very important in this regard, but also its proper design. That is because forging dies are usually subject to high cyclic stresses that can easily lead to a high level of internal pressure, local stress concentrations, localized deformation and finally premature failure of dies.

*Corresponding author, e-mail: dehghani@aut.ac.ir

Selecting a die material to satisfy all the properties required for a long life is not an easy task. Among the suggested materials, tungsten carbide has been widely used as the best choice [1–4]. However, there is a major concern remaining with this material, that is the lack of a long fatigue life. As tungsten carbide is a high strength, brittle material, the short fatigue life of dies might somehow be attributed to improper die design. That is because the high tensile stresses generated in the most critical regions of a die can readily reduce the fatigue strength, resulting in premature die failure. In other words, the importance of an appropriate die design to prevent or reduce the concentration of tensile stresses is just as important as die materials themselves.

It is reported that the decrease in the fatigue strength of dies, due to the presence of undesirable tensile stresses, can easily lead to the fracture of dies at a life shorter than expected [5–7]. As mentioned earlier, the main origin of detrimental tensile stresses is inadequate die design, resulting in undesirable stress concentrations. A high stress concentration leads in turn to crack initiation and fast crack growth, especially when the die material is brittle. This means that even if the best materials are selected, premature die failure can occur if a die is not designed properly.

Although a lot of papers have been published on the selection of suitable materials for dies [1–4], far fewer are available on the finite element modelling of die design. For example, reducing or eliminating the detrimental tensile stresses concentrated in the critical areas of dies is essential for improving the fatigue life of dies. Thus, the objectives of the present work were to determine and analyze the regions of highly concentrated stresses in forging dies, especially in the particular case of a bolt die. Then, attempts have been made for finite-element simulating the design of forging dies. To meet these goals, the practical approaches were studied, including die inserts, shrink rings, assembled dies and tapered dies. These alternative designs were investigated with regard to their effect on eliminating the tensile stresses concentrated in critical regions of a bolt die. The aforementioned techniques can significantly reduce the detrimental tensile stresses by imposing the pre-stresses leading to the prevention of premature failure of dies.

2. Experimental

Materials. Plastic deformation is not accepted to happen in the die because its occurrence in a die results in losing the dimension stability or size accuracy of produced bolts. In such a case, different bolts having various sizes may be produced with a large tolerance in dimensions. Consequently, tungsten carbide, which exhibits elastic behaviour rather than elastoplastic one due to its very high strength is considered for the die insert. Therefore, the material to be forged was considered AISI 1010 steel with elasto-plastic behaviour. The stress-strain relation for this steel is as follow [2]:

$$\sigma = 730\varepsilon^{0.22} \tag{1}$$

As for the die material, the characteristic parameters for tungsten carbide are, according to [3], as follows: E = 540 GPa, ν = 0.22.0. The result of the present simulation, applied for producing a M10 bolt made of AISI 1010, yielded the pressure of about 650 MPa.

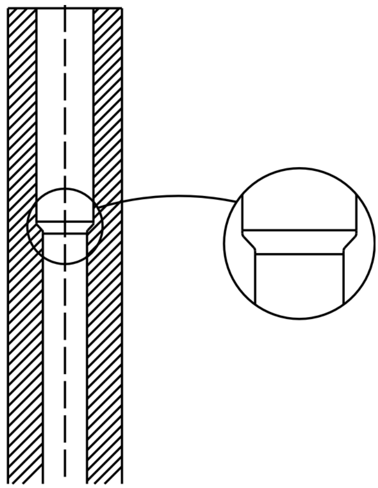

Fig. 1. Schematic of the design of an M10 bolt die

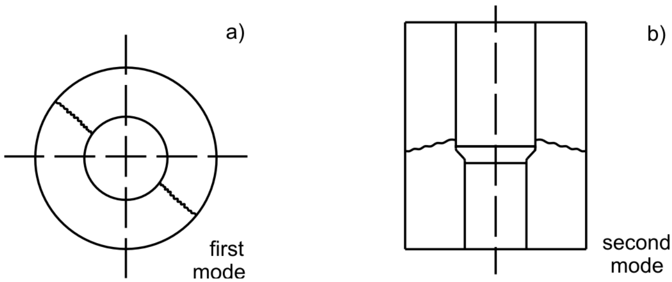

Fig. 2. Modes of possible failure due to: a) hoop stress and b) axial stress

A schematic design of the M10 bolt die studied in the present investigation is shown in Fig. 1. The design is based on existing standards [1, 2]. As any change in the cross section can cause stress concentration, therefore, in this case, the most significant fatigue failure is expected to occur at or near the transition section shown in Fig. 2. Due to the stress concentration, there are two possible origins for fatigue crack nucleation, as shown in Fig. 2. Contrary to the fact that the real life of forging dies should lie in the high cycle fatigue (HCF) range, here the premature fatigue failure is expected to occur in the low cycle fatigue (LCF) range. In other words, the undesirable tensile stress distribution in the die is responsible for this contravention and premature failure. As shown in Fig. 2, these tensile stresses are oriented in two directions; Figure 2a refers to the failure due to hoop stresses (first mode), whilst Fig. 2b indicates the failure due to axial stresses (second mode). This is also in agreement with the previous works [6, 8].

This work is mostly focused on the methods used to eliminate the tensile stresses in the die. This technique is in accordance with the work of others [5, 9]. The proposed approach is based on the following steps:

- Analyzing the stresses distributed in both die and work-piece using the finite element method.
- Determining the critical points or regions of highest stress concentrations and/or of maximum tensile stresses in the die.

A crude representation of a die as a thick cylinder is sufficient for some mathematical models, but it is not an adequate for accurately identifying critical regions of fatigue stresses. As a result, advanced numerical methods, especially the finite element methods, were used in this regard [10, 11]. It should be emphasized that in the present work, the ABAQUS software was used for finite element simulation [12].

The assumptions made in the finite element modelling were: both die and workpiece were analyzed simultaneously, the rigid behaviour of the die during the workpiece deformation was ignored, the symmetrical condition was considered, and the boundary condition was considered as 15 mm vertical displacement of a hammer pin.

3. Results and discussion

3.1. Stress analysis

Although the die material (i.e., tungsten carbide) yields a high tensile strength, its fatigue strength is considerably reduced due to the tensile stress generated across the die during the forging cycles. Thus, as a first step, it is essential to determine the distribution of tensile stresses in the die. The regions of high tensile stresses are the so-called critical points of the die. To determine these regions, a simple die, without shrink ring, was considered. Figure 3 presents the axial stress distribution within the different portions of the die during forging, whereas Fig. 4 shows the distribution of hoop stresses, both obtained using the assumed models. As is obvious, the hoop stresses in most portions are positive (tensile) and reach the maximum level across the region α of the die (Fig. 4). With respect to axial stresses (Fig. 3), the situation is different; they are positive or tensile in a much fewer number of] regions. By contrast, in most places, the generated stresses are negative (compressive).

It is interesting that at the critical radius, i.e. at the point of cross-sectional change, where crack initiation is most likely to occur, the axial stresses are tensile (region β indicated in Fig. 3). Thus, these two regions (α and β) can be identified as the critical zones where the stress concentrations are the highest. That is why the first mode of fracture (Fig. 2a) can take place in the region α, while the second mode of failure (Fig. 2b) may occur in the region β.

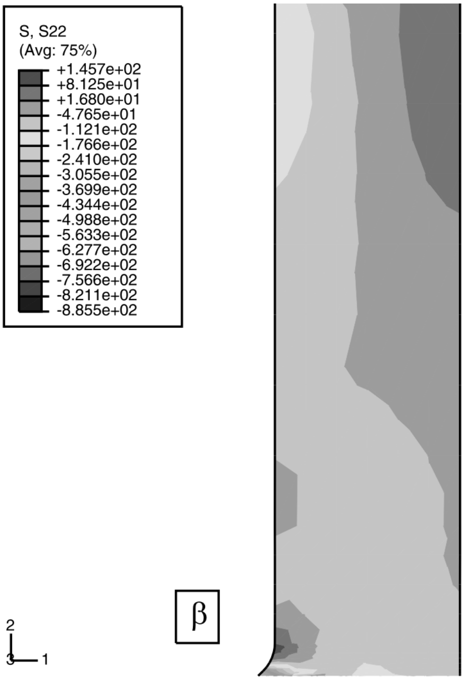

Fig. 3. Axial stress counter in the insert (MPa)

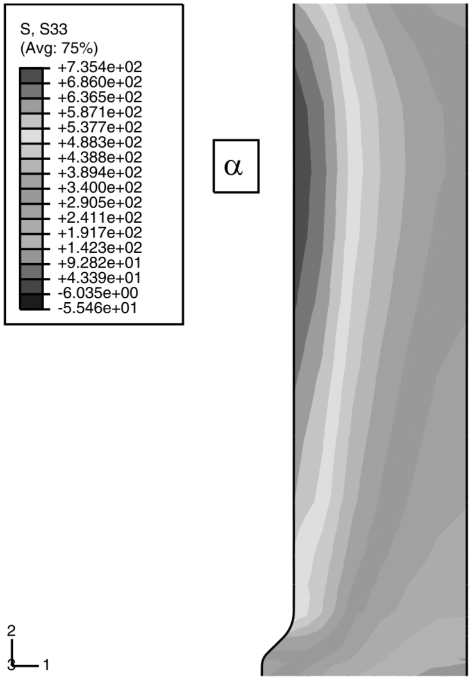

Fig. 4. Hoop stress counter in the insert (MPa)

Therefore, according to the fatigue analysis, the fatigue strength is reduced in the regions having positive stresses. As a result, premature fatigue failure is expected to occur anywhere in the die where the generated tensile stress is at its highest value [13, 14]. That is why the lifespan of forging dies is mainly considered to lie in the low-cycle fatigue (LCF) range rather than in the high-cycle fatigue (HCF) one. Nevertheless, as already mentioned, if these principal stresses could be changed to compressive stresses (negative), e.g. by using a shrink ring to induce pre-stresses, then a significant increase in the fatigue strength of dies could be readily attained. If however the tensile stresses, dominant during forging cycles, could not be eliminated, reduced, or replaced with compressive ones, then there is no alternative to using a tougher material.

3.2. First mode of failure caused by hoop stresses

In this section, the analysis and control, by means of FEM simulation, of the first mode of failure, as illustrated in Fig. 2a, caused by hoop stresses is discussed. One of the best approaches for converting the hoop tensile stresses (generated during cyclic forging) to compressive ones is use a shrink ring, which is simulated here. In such a case, there is a pre-stress imparted by the ring to the inner part of the die. This in turn causes the die to experience an overall negative hoop stress, even under the maximum inner pressure imposed on the die insert during service.

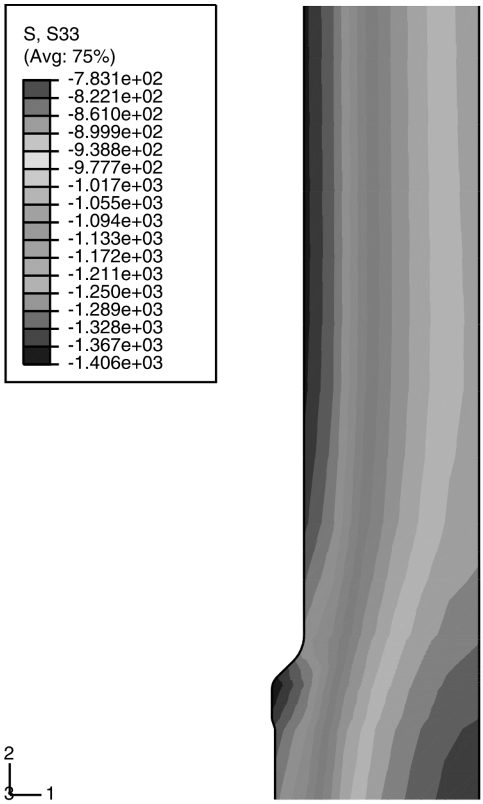

Fig. 5. Hoop stress counter imposed only by using shrink ring (MPa)

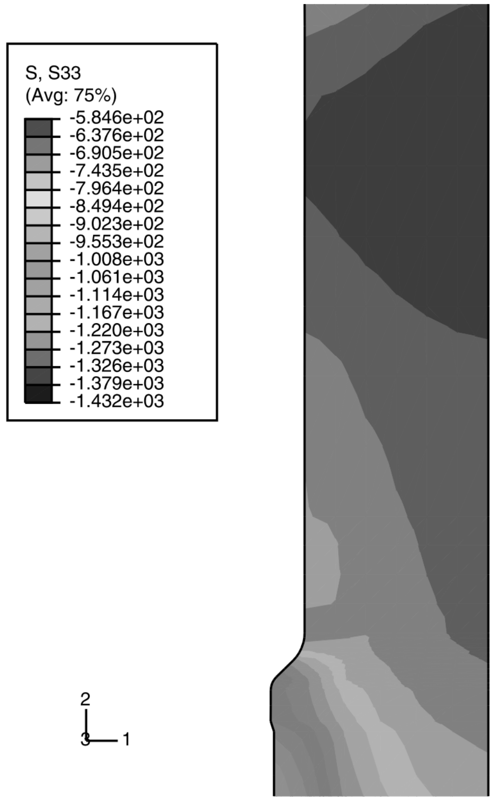

Fig. 6. Hoop stress counter when applying load along with using shrink ring (MPa)

Classical analysis, based on Lame's method [15], shows that it is possible to simulate and calculate the interference of a shrink ring. To illustrate the effect of a shrink ring, the results of FEM simulation regarding its hoop-stress counter are compared in Figs. 5 and 6. Figure 5 appertains only to the presence of the ring, but not to finer attributes, such as the inner pressure in the absence of loading. Figure 6 shows the simulated hoop stresses induced by both the ring load and the forging load. As is obvious, in both cases, the resultant hoop stress in all parts of the die is negative. As a result, the fatigue life of the die can be considerably increased if the first mode of die failure can be delayed by the use of shrink rings.

3.3. Second mode of failure caused by axial stresses

As previously mentioned, another cause of die failure (second mode) is due to the generation of tensile stresses in the transition region of the die, (Fig. 2b). In this case, three techniques may be applied to overcome the premature failure of the second mode. The first one is to use a multipart die, i.e. an assembled die. A schematic diagram of a possible, assembled die for the M10 bolt is shown in Fig. 7. Here, the assembled die consists of only two parts. However, in the case of a complicated shape, a more elaborate assembly might consist of, for example, several parts, such as a case, sleeve, middle insert, back insert and front insert. In this procedure, the interface between various die parts may act as free surfaces that relieve the tensile stress, thereby reducing the stress concentrations at or near the critical sections. This is an important practical issue for increasing the fracture toughness of materials when a thick structure experiencing plane strain condition with low toughness is divided into thin layers experiencing plane stress condition with high toughness [16]. In the case of dies, this happens when they are manufactured of various interferences, e.g. an assembled die, which consists of a few parts or sections.

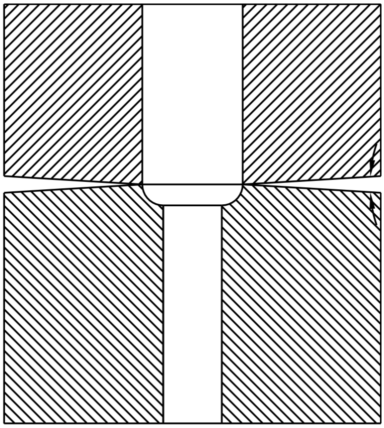

Fig. 7. The schematic of an assembled M10 bolt die consisting of two parts

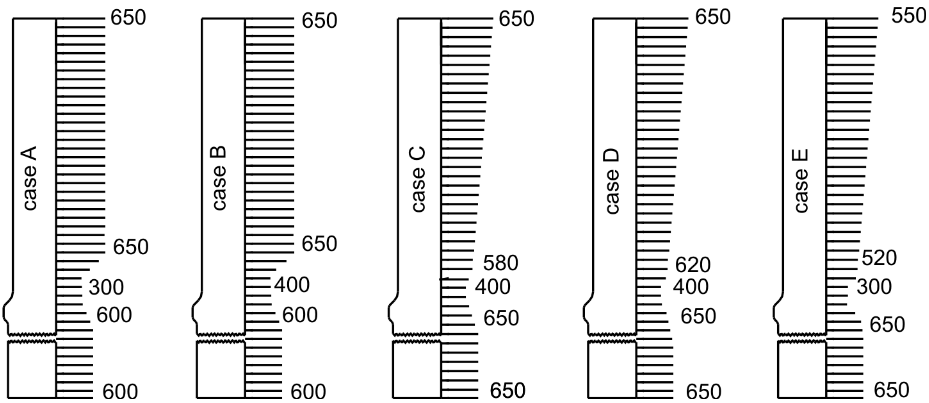

Fig. 8. Five different pressure distributions across the die wall (MPa)

The other method, already mentioned, is to impose a pre-stress (i.e., negative stress) across the die wall, especially at or near the cross-section change-zones or the critical transition zones. To model this case, five situations (A, B, C, D and E) of nonuniform pre-stresses imposed on the die are assumed, and presented in Fig. 8. As the primary simulation, applied for producing a M10 bolt made of AISI 1010, yields the pressure of about 650 MPa, therefore, the upper limit of simulated pressure was considered 650 MPa,

For the cases C, D and E, the presumed pressure is reduced linearly from a maximum at the top, toward the transition section, and then remains constant at the bottom part of the bolt die. However for cases A and B, the pressure is reduced only at the transition section, while at both the top and bottom portions of the bolt die, it is kept constant. The reason for simulating such a pressure distribution was to generate a kind of negative bending moment, to counteract the positive one produced by the forming pressure, as shown in Fig. 9a. It should be emphasized that the simulated pressure distributions shown in Fig. 8 are obtained by the finite element method as well as by the ABAQUS software.

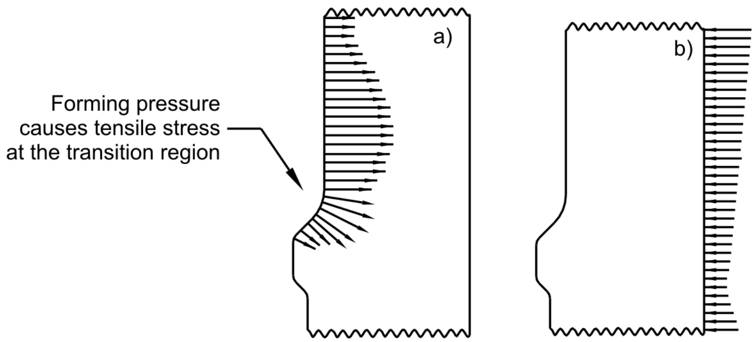

Fig. 9. Forging pressure that generates a bending moment (a) and imposed pressure that can eliminate bending effects (b)

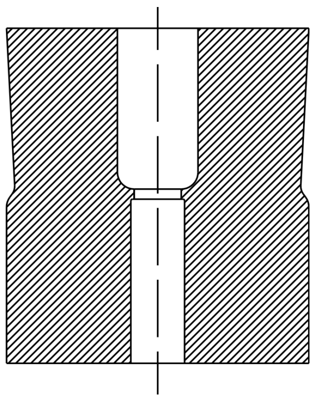

Fig. 10. Scheme of the proposed tapered die

In general, the forging pressure generates a positive bending moment that in turn causes positive tensile stresses at the cross-section change, as shown in Fig. 9a. These tensile stresses can readily result in fatigue crack initiation, leading to the premature fracture of the die. That is because the generation of detrimental bending moments can result in the second mode of failure, as presented in Fig. 2b. As mentioned before, one useful technique to counteract these undesirable bending moments or tensile stresses is to employ one of the five pre-stresses described in Fig. 8, illustrated again in Fig. 9b to show the mechanism of compensation of bending effect. Such a condition can be created using a tapered insert. For instance, an alternative die design for an M10 bolt using a tapered insert is shown in Fig. 10.

3.4. Numerical analyses

As already mentioned, FEM was used to simulate the forging process for the tapered die design proposed in Fig. 10. As the overall stress should be negative in the critical regions, to prevent the premature failure, attempts were therefore made to identify the critical elements susceptible to crack initiation.

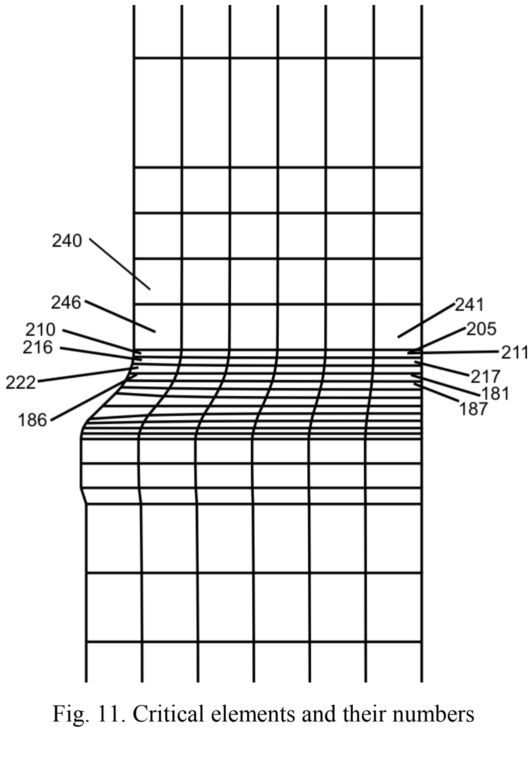

Fig. 11. Critical elements and their numbers

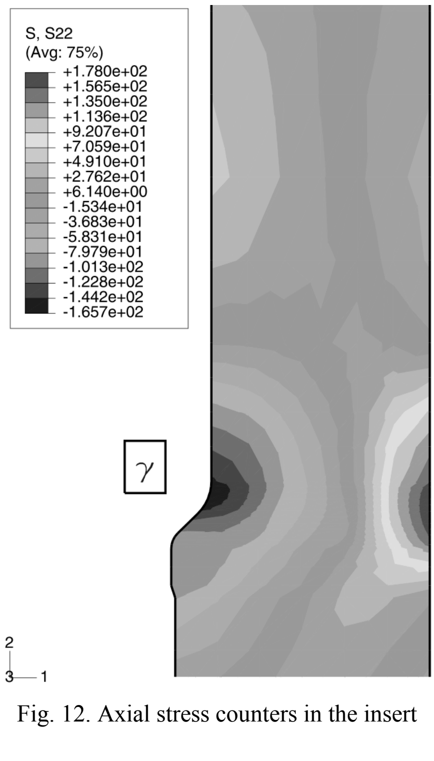

Fig. 12. Axial stress counters in the insert

Figure 11 shows the number of elements in the finite element simulation located at critical zones. Interestingly, for the different cases of A to E, illustrated in Fig. 8, the hoop stresses in all critical regions of the die insert, including the critical fillet region β, remain negative during a complete forging cycle. On the other hand, as shown in Fig. 12 and referring to the results summarized in Table 1, the axial stress at the critical fillet region β remains negative during a complete cycle. This confirms the validity of the proposed approach in imposing the proper pre-stresses to generate negative stresses at critical zones of the die.

Table 1. Axial stresses in the critical elements in the fillet region (MPa)

Element number										Case
240		246		210		216		222		
After loading	Before loading	After loading	Before loading	After loading	Before loading	After loading	Before loading	After loading	Before loading	
–191	–135	–134	–177	–67	–203	–62	–208	–104	–208	A
–198	–114	–136	–157	–67	–185	–62	–191	–102	–193	B
–191	–103	–127	–136	–56	–159	–51	–164	–92	–165	C
–183	–114	–115	–151	–42	–177	–36	–182	–78	–183	D
–215	–141	–142	–167	–64	–183	–55	–185	–96	–182	E

As is obvious, due to this design, the imposed stresses on the die insert in the critical fillet region are negative. This again can lead to longer fatigue life of dies, by eliminating the undesirable tensile stresses. Therefore, generating negative or compressive stresses is the best method to increase the fatigue life of components. These stresses are typically produced by surface treatments such as shot preening, cold work, heat treatments, plating, etc. [16]. Generating the compressive stresses is practical not only for dies but also for other important rotating parts, including axles, shafts, crankshafts, etc. that are subjected to cyclic loading. However, there is a region in Fig. 12, labelled γ, where a positive axial stress is generated as well. Indeed, this region must be inspected and treated with care when being subject to forging cycles. The numerical simulations reveal that the axial stress varies from positive to negative in this region during a loading cycle.

In another attempt, the mean stress σ_m and alternative stress σ_a were simulated. The values for σ_m and σ_a, obtained using the finite element analysis for the critical elements numbered in Fig. 11, are summarized in Table 2. These elements are analyzed for five cases illustrated in Fig. 8. In general, the lower the σ_m and σ_a, the higher the fatigue life will be. On the other hand, it should be considered that it is impractical to lower both σ_m and σ_a for a process like forging.

The generation of cyclic stresses in the elements located in region γ in Fig. 12 enhances crack initiation at this region. That is because, due to the generation of tensile stress, this region is the area most susceptible to failure. As a quick approach, the values of σ_m and σ_a appertaining to the critical elements, can be used to determine the most critical case among the five cases, A–E, illustrated in Fig. 8. For instance, it can be concluded that by comparing the five cases, the optimum values, in terms of σ_m and σ_a, are probably those corresponding to cases B and C.

Table 2. Mean and alternative stresses in critical elements (MPa)

Element number										Case
205		211		217		181		187		
σ_a	σ_m	σ_a	σ_m	σ_a	σ_m	σ_a	σ_m	σ_a	σ_m	
44.5	153.5	44	154	45	152	45.5	149.5	47.5	145.5	A
37.5	145.5	37	149	38	150	39	149	41	146	B
31.5	133.5	32.5	138.5	34.5	140.5	36.5	140.5	39.5	138.5	C
46	131	49	135	51.5	135.5	53.5	133.5	57	132	D
37.5	151.5	38.5	152.5	40.5	152.5	42.5	149.5	50	146	E

On the other hand, another approach might be used to estimate the length of the life cycle for various stresses occurring in critical elements. The method enables determining the most critical element and finding the best case of A–E as well as estimating the life cycle numerically. In general, the fatigue life can be estimated from [17]:

$$N=\left(\frac{S_f}{a}\right)^{1/b} \tag{2}$$

where a and b are the constants defined below, and S_f is the fatigue strength for a specific life

$$a=\frac{\left(f S_{ut}\right)^2}{S_e} \tag{3}$$

$$b=-\frac{1}{3}\lg\left(\frac{f S_{ut}}{S_e}\right) \tag{4}$$

where f is a constant in the range 0.8–0.95, S_{ut} is the ultimate strength and S_e is the endurance strength of the material. On the other hand, according to Goodman's theory, the relation between the fatigue parameters is as follows [17]:

$$\frac{\sigma_a}{S_f}+\frac{\sigma_m}{S_{ut}}=1 \tag{5}$$

Substituting Eq. (5) into Eq. (2) leads to the following formula for determining the fatigue life or the number of cycles N:

$$N=\left(\frac{\sigma_a S_{ut}}{a\left(S_{ut}-\sigma_m\right)}\right)^{1/b} \tag{6}$$

The values for σ_m and σ_a (Table 2) are obtained using FEM simulation. The assumed values for the ultimate and the endurance strengths were 800 MPa and 350 MPa, respectively. The value of 0.9 was used for the constant *f*. Now, the number of cycles to failure can be calculated.

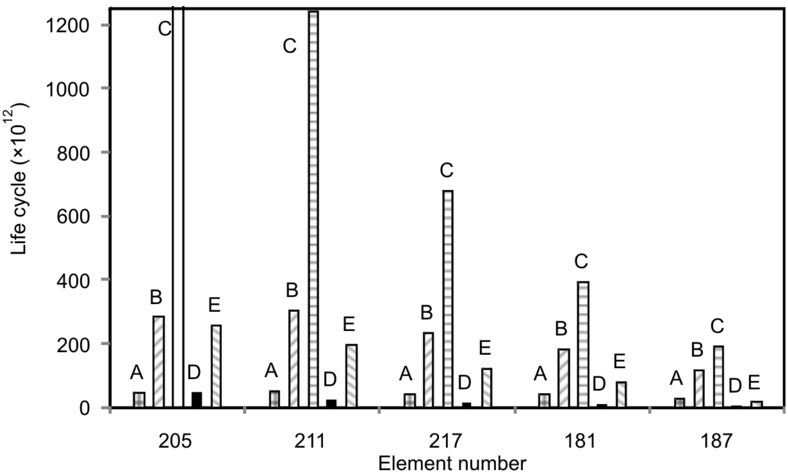

Fig. 13. Life cycle in accordance with cases A to E pertaining to critical elements

In Figure 13, the number of cycles to failure (the fatigue life) obtained for cases A–E are shown, pertaining to the critical elements subjected to cyclic positive/negative loading. The best case among A–E, the most critical element and the final life cycle can be readily determined by comparing the numbers of cycle to failure for various situations. To find the best case among the cases A–E, the number of cycles or fatigue life obtained for all states are compared in Fig. 14.

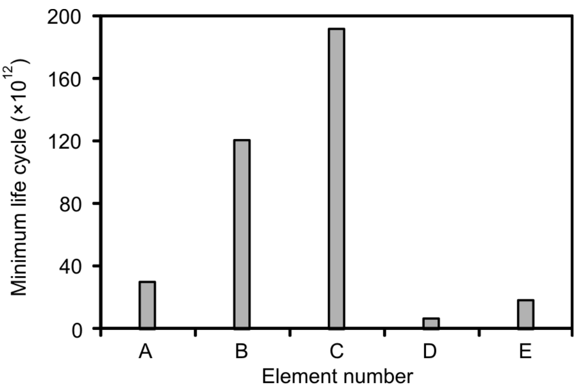

Fig. 14. Minimum life cycles for cases A–E

Obviously, the order C, B, A, E and D can be considered as the best to the worst case in terms of fatigue life. In other words, case C has the longest fatigue life because it has the highest number of fatigue cycles (1.9×10^8 cycles). Besides, even for a given case, the most critical element can be determined in terms of the number f of cycles to failure. For example, referring to Fig. 13 and comparing the fatigue life of different elements in case C, the failure will most probably initiate at the element 187 which has the lowest cycle to failure.

From the industrial point of view, the application of numerical simulation, e.g. FE analysis, not only provides precise analysis of stresses, accurate calculation of forming loads and rapid solutions to technical problems , but it also saves costs and time.

4. Conclusions

The possible ways of improving the fatigue life and preventing the pre-mature failure of forging dies have been investigated, by using techniques such as imposing pre-stress by shrink rings, die inserts, shrink rings, assembled dies and tapered dies. The studies were carried out on a M10 bolt die made of AISI 1010 steel. The results of finite element simulation show that negative or compressive stresses are generated using the aforementioned approaches. In other words, the methods applied can reduce significantly the tensile stresses responsible for fatigue crack initiation. The fatigue life of dies can be significantly improved when the negative stresses, rather than tensile ones, are formed. In another approach, the critical regions of highly concentrated stresses are identified and appropriate solutions are provided. For example, changing the design of a die was considered as a practical approach that has significant influence on the fatigue life of dies.

References

[1] LANGE K., *Handbook of Metal Forming*, McGraw-Hill, New York, 1985.
[2] ALTAN T., OH S.I., GEGEL H.L., *Metal Forming: Fundamentals and Applications*, American Society for Metals, USA, 1983.
[3] HUR K.D., CHOI Y., YEO H.T., Finite Elem. Anal. Des., 40 (2003), 173.
[4] GRRAY A.G., *Source Book on Cold Forming*, American Society for Metals, USA, 1975.
[5] VAZQUEZ V., HANAN D., ALTAN T., J. Mater. Proc. Technol., 98 (2000), 90.
[6] YOH E.G., KIM Y.I., LEE Y.S., PARK H.J., NA K.H., J. Mater. Proc. Technol., 131 (2002), 647.
[7] COLLINS J.A., *Failure of Materials in Mechanical Design*, Wiley, New York, 1981.
[8] FALK B., ENGEL U., GEIGER M., J. Mater. Proc. Technol., 119 (2001), 158.
[9] KNOERR M., RENTSCH C., ALTAN T., *Application of 1-DEAS in CAE of Forging Dies*, Proc. ICCON User's Conference, OH, USA, 1991.
[10] YEO H.T., CHOI Y., HUR K.D., Int. J. Adv. Manuf. Technol., 18 (2001), 54.
[11] LONG H., BALENDRA R., J. Mater. Process. Techn., 81 (1998), 665.
[12] ABAQUS User's Manual. Version 6.3.1, Hibbitt, Karlson and Sorensen, Inc., USA, 2002.
[13] FUCHS H.O., STEPHENS R.I., *Metal Fatigue in Engineering*, Wiley, New York, 1980.
[14] GEIGER M., FALK B., ENGEL U., Prod. Eng., 6 (1999), 2.
[15] UGRAL A.C., FENSTER S.K., *Advanced Strength and Applied Elasticity*, Prentice-Hall, New Jersey, 1995.

[16] HERTZBERG R.W., *Deformation and Fracture Mechanics of Engineering Materials*, Wiley, New York, 1983.
[17] SHIGLEY J.E., MISCKE C.R., BUDYANS R.G., *Mechanical Engineering Design*, McGraw-Hill, New York, 2004.

Received 29 December 2008
Revised 12 October 2009