Indian Journal of Engineering & Materials Sciences
Vol. 23, August 2016, pp. 239-246

Fatigue life modeling of gear like products using ANN

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Received 2 September 2015; accepted 2 August 2016

The expected life of a gear is important parameter especially for the gears to secure of the mechanics of it. Related to this importance, fatigue failure is one of the most seen failures occurred on gears working under cyclic loads. It is not possible to eliminate fatigue failure effects but it is possible to reduce by appropriate materials selection and design criteria. Due to demand for gears with higher load-carrying capacity and increased fatigue life, it is important to determine the fatigue strengths of the gears. In this study, forward extrusion method with cosine and tapered profile dies is carried out to obtain gear-like products. The products are then tested under cyclic loads to determine the fatigue life. The results obtained from the experiments are used as inputs in developing the ANN models. Different ANN models are developed for cosine curved and straight tapered profiles to obtain the best models. A comparative analysis is performed in order to evaluate the accuracy of the developed models, in terms of statistical measurements (R^2, MSE, MAE). Results revealed that proposed ANN models for both cosine curved and tapered profiles are able to predict the fatigue life of the gear-like profiles.

Keywords: fatigue, Gear forming, ANN, Extrusion

As defined by the AGMA gears are machine elements that transmit motion between axis and plane. In the design of a machine element which subjected to cyclic loading such as gears there are several requirements that must be met. Engineers have been working for years on the development of advanced materials and manufacturing methods, in order to design gears with stronger teeth especially the failure problem of gears. There are a lot of different material and design parameters to consider when determining the number of loading cycles needed for fatigue crack to appear. As known, two types of stresses can change in gear pair in the course of power transmission: (i) bending stress and (ii) surface contact stress. On the other hand, two types of teeth damage can occur on gear when it is subjected to cyclic loads: gear pitting and tooth breakage[1]. The determination of the maximum stresses in a loaded gear tooth is complicated by the variation in magnitude and direction of the load on the tooth during contact and by the shape of the tooth, since it has varying width and is joined to the body of the gear by a fillet[2-4]. In general, the strength of a tooth gradually increases from the tip to root of the tooth.

Many studies have been carried out on the topic of fatigue problem of gears. Sraml and Flaskel[5] used Hertzian theory for the computational modeling of contact fatigue damage initiation. They also used FEM to specify the number of stress cycles needed for the primary fatigue damage to happen. Fredette and Brown[6] attempted to reduce the stresses around a gear root tooth by drilling holes along the axis of a gear segment. Results revealed that it is possible to improve the strength and durability of a gear with the addition of design features in critical areas. Sankar and Nataraj[1] investigated the change in bending strength of a spur gear by replacing its trochoidal fillet with a circular root fillet. Finite element simulations were conducted on standard and modified spur gears using ANSYS software in purpose to determine the change in the strength of tooth as well as minimize the failure in the spur gear. A stress-strain analysis in the framework of FEM was conducted by Krumberger *et al.*[7] in order to analyze the gear crack propagation. The process was divided into two periods; crack initiation and crack propagation. Simulations were done by variable loading in order to determine the life of the spur gear. In another study, Krumberger *et al.*[8] determined the cycles needed for fatigue crack in a thin-rim gear to appear on the basis of FEM analysis. Mohanty[9] has used analytical method for the calculation of the load sharing of HCR spur gears. A detailed investigation on contact stresses in spur gears has been carried out by

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Thirumurugan and Muthuveerappan[10] by conducting a single point loaded FEM model. The effect of different FE models and certain gear parameters were analyzed in order to determine

the load sharing ratio and the stresses. Li[11] used FEM to investigate the tooth contact strength of spur gears for different contact ratios and addendums. Contact analyses were conducted based on the LSR (load shearing ratio) using a mathematical programming method.

In practice, gears usually work under dynamic loads. Because of this, a localized damage can cause a tooth breakage, which most often leads to a total gear failure. The failure analysis based on stress-life diagram is most widely used method for gear design applications. Fatigue life diagrams can be used to show the behavior of a material under various conditions. Obtaining such information requires a large number of experiments. Artificial Neural Networks (ANNs) may be used in predicting fatigue life of gears beforehand thus reducing the number of experiments needed. The use of ANN in predicting a wide range of material properties and forming methods has been investigated by several researchers. These researchers have used ANN models to estimate, forming loads[37-38], fatigue life diagrams[12-20], flow curves[21-27], as well as bending properties[28-33], wear loss and surface properties[34-36] of various types of materials and methods. Genel[15] attempted to predict the fatigue life properties of steels using tensile material data. It was pointed out that it is possible to make fairly good predictions using ANN. Lee *et al.*[18] developed an ANN model for the estimation of the fatigue damage based on the experimental results. Results revealed that by use of ANN, fatigue damage can be estimated with relative ease. Mohanty *et al.*[19] applied ANN for fatigue life estimation of two different aluminum alloys. It was observed that predicted results for both alloys were in agreement with the experiments. Junior *et al.*[20] demonstrated the generalization capability of ANNs in building life diagrams by using a few S-N curves.

In the light of the above mentioned studies, it can be seen that the use of ANN in modeling of various types of materials and forming methods became widespread. However, it should be noticed herein that, to the authors' best knowledge, a predictive ANN model for fatigue life prediction of gear like products has not been studied in the literature and is the subject of current study. In this study; the cyclic loads were applied on the products which were produced by forward extrusion with cosine and tapered profile dies with different die land lengths to determine the fatigue strength. The experimental fatigue stress values of each product were measured and recorded. Later on, possibility of using ANNs for the lifetime predictions of gear like products was investigated by conducting networks with different configurations using the data derived from the experiments that were carried out using different sample types. The excellent capabilities of ANNs in fitting the experimental stress-life curves were also evaluated.

Methodology

In this study, gear like profiles were produced by forward extrusion with two different die profiles; cosine curved and straight tapered. Samples were tested under variable stresses and experimental results for selected parameters were recorded per sample giving a total of 96 data to analyze. Using the data available, ANN models were developed for fatigue life prediction. Then, ANN results were compared with the experimental results. Having sufficient data available can be particularly helpful in understanding how well the network is performing. Flowchart for the present study is given in Fig. 1.

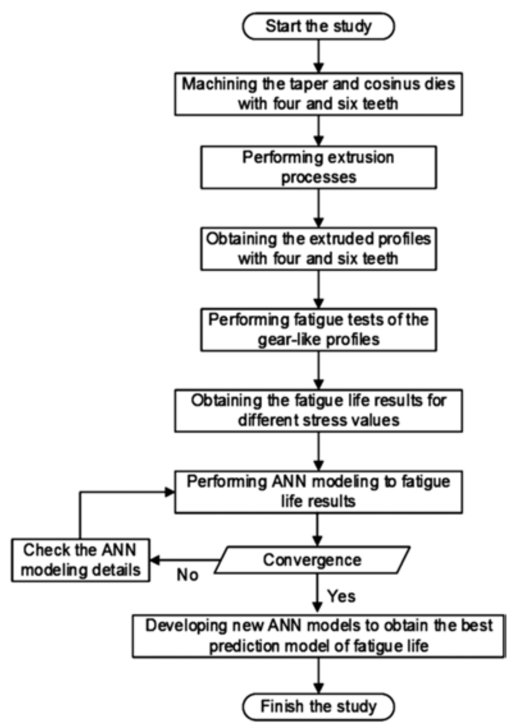

Fig. 1—Flowchart of the study

Experimental study

Al1050 material was used in the experiments. The chemical compound of Al 1050 was given in Table 1.

The stress-strain relationship of AA 1050 which is a very important mechanical property of the material was determined from compression test. It is given as;

$$\sigma = 138\varepsilon^{0.156} \text{ MPa} \quad \ldots (1)$$

Billets were cut into sections from the bar, each 45 mm in length. Then, the diameter of the billets was machined to 28 mm. The container has an inner diameter of 28.2 mm and outer diameter of 60 mm. The punch was made from the same material and has the diameter of 28 mm.

Two different transition geometries, cosine curved and straight tapered dies, with four and six teeth profiles were used in the present study. Punches, containers and billets were cut out using CNC machine. The dies were made by wire-cut EDM machine due to their geometrical complexity. Dies and other tools were made from 1.2344 DIN hot worked tool steel and hardened to 54 HR_c. Photographical view of the experimental set up were given in Fig. 2. The hydraulic press used in the experiments is a 150 metric ton press having a constant ram speed of 5 mm/s. Before the experiments, all the billets were cleaned using acetone in order to provide the similar friction conditions. Photographical view of the extruded part can also be seen in Fig. 2. Cosine and straight tapered die transition profiles with three different die land length which were 15, 20, 25 mm. respectively were used in the forward extrusion experiments.

Table 1—Chemical compound of AA 1050 aluminum.

Si	Fe	Cu	Mn	Mg	Zn	V	Ti	Pb	Al
0.158	0.291	0.012	0.005	0.012	0.018	0.01	0.008	0.005	Balance

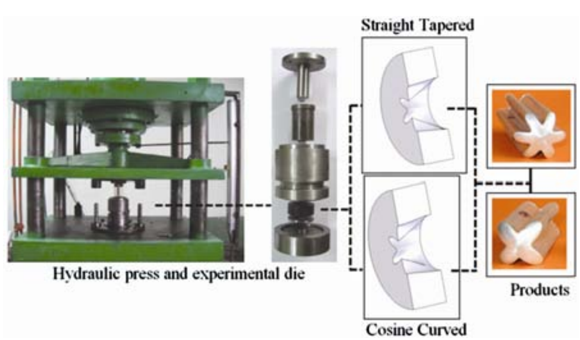

Fig. 2—Experimental die setup and extrusion products

INSTRON 8501 Universal Test Machine was used for the experiments in this study. Upper and lower apparatus were machined from 1.2344 steel, hardened in oil and tempered to 52R_c for fatigue tests. Experimental setup of fatigue tests can be seen in Fig. 3.

Loading effect on the machine parts is related with the location of the applied force. Total bending load for gears occurs in the dedendum. Resultant stress (W_{bend}) value is given in Eq. (2);

$$W_{bend} = \frac{I}{c} = \frac{y * h_{min}^{2}}{c} \quad \ldots (2)$$

and expressed in Eq. (3).

$$\sigma_{bend} = \frac{M_{e\breve{g}}}{W_{e\breve{g}}} \quad \ldots (3)$$

Moment value according to gear geometry can be expressed as in Eq. (4);

$$M_{bend} = 5.133 * F_t \quad (4)$$

So resultant fatigue bending stress on the gear can be written as:

$$\sigma_{bend} = \frac{(30 * F_t)}{y * h_{min}^{2}} \quad \ldots (5)$$

h_{min} and y values of each gear is measured and recorded for accurate calculations.

Load is given schematically in Fig. 4 and can be defined as F_t=F/2 for tooth and it is forced to be bent and fractured. Moreover, fatigue stress occurs at a critical point near dedendum, hence determination of the loading point has a great importance to analyze the experiments.

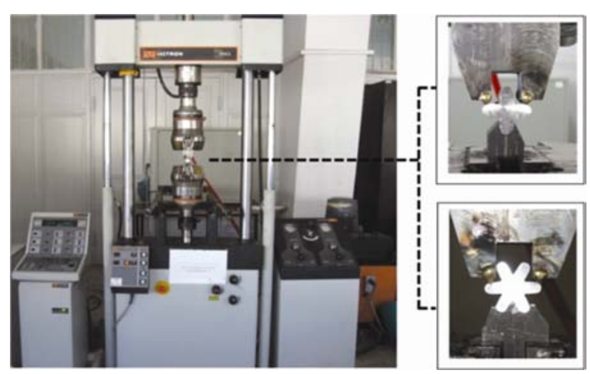

Fig. 3—Experimental setup of fatigue tests

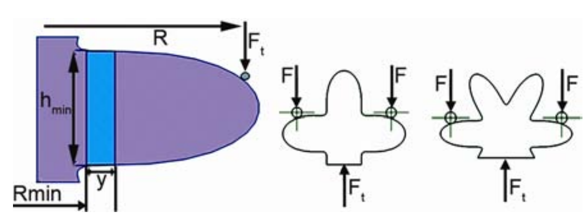

Fig. 4—Schematical view of loads on gears for fatigue test (a) detailed view of loading, (b) 4 teeth gear loading and (c) 6 teeth gear loading

Loading to gears is carried out by the movement of Instron 8501 Universal Test Machine's jaws. A force couple on the gears is manifested by means of an implemented F_t force. The distance between this force couples is calculated as 18.2 mm in 4 teeth gears and as 20 mm in 6 teeth gears. By opening two channels on the upper loading apparatus in the testing set two pins with 2 mm diameter are attached to these channels. In this way, the force couple is made to affect through a single distributed loading line on the gears sensitively.

Development of the ANN modeling

Different learning rules can be used in order to improve the ANNs' performance. Back propagation is a common algorithm for training the ANNs since it has the advantages of being very simple and accurate. After the network is initialized with random weights, the method continuously updates the weights to match the required output until the loss function is minimized.

The results obtained from the experiments were used for the development of the ANN models. The inputs of the network are tooth number (*N*), die bearing length (*L*) and stress (σ), and the only output is fatigue life (log*N*). The modeling was divided into two sections; the first ANN model is related to the fatigue life prediction of tapered die profile. The second ANN model is related to the estimation of fatigue life of cosine die profile. The structure of the developed neural networks is given in Fig. 5.

A transfer function is necessary to translate the input signals to output signals. The choice of the transfer function in ANNs is of great importance to their performance however there isn't a strict rule for selecting transfer function. The selection depends mostly on experience. In this study, two transfer functions, Tanh Axon and Sigmoid Axon, were used to introduce nonlinearity into the network. An ANN structure with the feed-forward neural networks was used to estimate the fatigue life of tapered and cosine gear. The total experimental data (48 samples for tapered profile and 48 samples for cosine profile) was randomized and divided into two categories named training subsets (80% of total data) and testing subsets (20% of total data). The ideal transfer function and the number of neurons and hidden layers should be found through a trial and error method. In the selection of best network structure, the measurements (R^2, RMSE, *MAE*) were used as the performance criteria between the ANN predicted and the experimental values. The effect of different network structures and training algorithms on the ANN models is presented in Table 2.

As shown in Table 2, the network structure with 12 neurons (Model 3) provided fairly good prediction results for the tapered profile and the network structure with 14 neurons (Model 9) produced the best

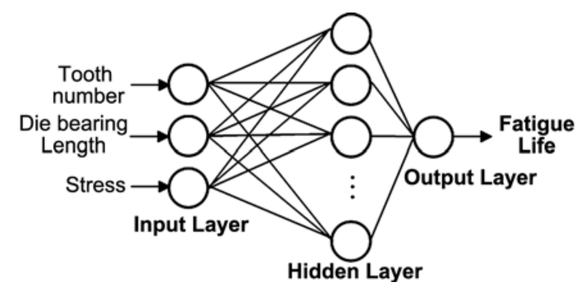

Fig. 5—Neural network structure for fatigue life estimation

Table 2—Statistical values of all the performed ANN models.

ANN Models of Tapered Profile	Algorithm	Function	Neuron Number	MSE	MAE	R^2
M1	TanhAxon	LM	4	0.135	0.318	0.678
M2	TanhAxon	LM	8	0.128	0.295	0.799
M3	TanhAxon	LM	12	0.032	0.153	0.946
M4	TanhAxon	Momentum	6	0.039	0.169	0.935
M5	SigmoidAxon	LM	8	0.065	0.215	0.898
M6	SigmoidAxon	Momentum	16	0.052	0.198	0.907
ANN Models of Cosine Profile	Algorithm	Function	Neuron Number	MSE	MAE	R^2
M7	TanhAxon	LM	6	0.036	0.124	0.898
M8	TanhAxon	LM	12	0.022	0.085	0.982
M9	TanhAxon	LM	14	0.002	0.030	0.999
M10	TanhAxon	Momentum	4	0.015	0.113	0.982
M11	SigmoidAxon	LM	8	0.034	0.114	0.901
M12	SigmoidAxon	Momentum	14	0.029	0.095	0.924

performance for cosine die profile in terms of statistical criterions (R^2, *MSE*, *MAE*). In particular, exceptional agreement was obtained adopting the LM algorithm. Furthermore, changes in the transfer function also impact the model estimation capabilities. In this regard, relatively better responses were provided by TanhAxon functions. Considering these results and disregarding small variations, it can be said that, as far as the fatigue life prediction is regarded, the models offer fairly strong performance.

Results and Discussion

Since the gears are subjected to variable stresses by general operating conditions, they carry out their duties under continuous fatigue effect. In this study, a test type which affects the gear and determines the resistance of the gear as a result of one-way bending fatigue procedure is used to determine the resistance of manufactured gear parts against the fatigue. The conducted experiment is extremely suitable with regards to determining the effects of parameters such as production method, material geometry and material to the resistance of the product. In order to determine the fatigue resistances of the gears, implemented stress values must be known or calculated. Implemented stress values on the gear in the conducted test are given in Table 3.

Effect of die transition profile and die land length

In order to compare fatigue strengths of 4 and 6 teeth parts, implemented stress values were equal in the experiments for each test. At the moment any damage occurred on the gears the experiments were stopped, number of cycles is noted and then following Wohler curves were obtained for each product separately and diagrams were concluded and the best fatigue resistant die set was chosen for which die land length is 15 and die transition profile is cosine for both 4 and 6 teeth gears.

Table 3—Applied stress and load values of fatigue tests

	4 Teeth gear	6 Teeth gear
Stress (MPa)	Load (kN)	Load (kN)
76	-1.75±1.5	-0.950±0.700
92	-2.125±1.875	-1.125±0.875
98	-2.375±2.125	-1.250±1.000
115	-2.625±2.375	-1.375±1.125
126	-2.875±2.625	-1.500±1.250
138	-3.125±2.875	-1.625±1.375
182	-4.125±3.875	-2.125±1.875

4 teeth products were obtained by using dies having cosine dies with 15, 20 and 25 mm die land length and experimented on fatigue test. When graphics were examined together, it can be seen that fatigue strength values are extremely close to in each other in logarithmic scale. By the increase of die land length the fatigue strength decreases even if by a little.

Experiments were conducted by using also 4 teeth but tapered dies having different die land lengths and selecting appropriate stress values. When the results were examined, very close results attract the attention. However, it can also be seen that parts with shorter die land lengths have a little bit better fatigue strength.

It was determined that extrusion products of 4 teeth parts' fatigue strength values vary according to die land length and transition profile. At the same die land length values, products which were manufactured with dies having cosine die profile have more strength than products produced with die products having tapered die profile. It was observed that die transition profile in these dies produces more deformation than tapered dies and in this case the product's fatigue strength increases. It is clear from the results that best fatigue resistant product is obtained from the cosine curved die with 15 mm die land length and its fatigue stress-cycle curve is seen in Fig. 6. Only one diagram was given in the study because of the similarity of the results in the logarithmic scale hence detailed fatigue stress-cycle results were given in Table 4.

Likewise, 6 teeth products were produced by forward extrusion method. Fatigue strengths of

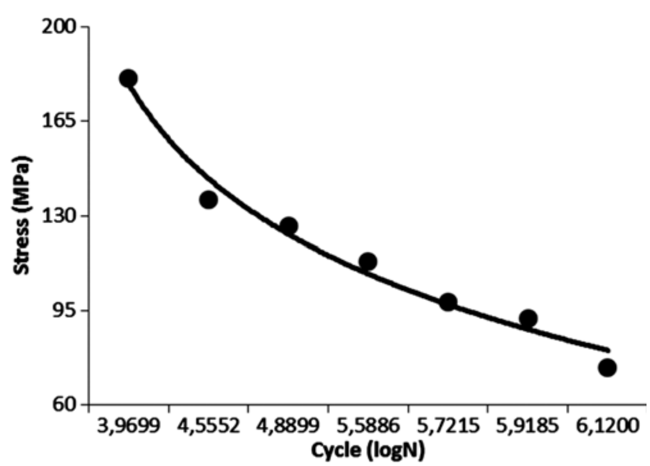

Fig. 6—Fatigue stress-cycle curve of 4 teeth cosine profile for 15 mm die land length

Table 4—Stress-cycle values of the fatigue tests

Stress (MPa)	Cycles (logN)					
	6 Teeth Cosine 15 mm	6 Teeth Cosine 20 mm	6 Teeth Cosine 25 mm	6 Teeth Tapered 15 mm	6 Teeth Tapered 20 mm	6 Teeth Tapered 25 mm
182	3.36654	3.33746	3.32139	3.35469	3.29973	3.27646
138	4.47083	4.44567	4.41191	4.43908	4.41948	4.38672
126	5.12237	4.98875	4.88995	4.77459	4.74856	4.85546
115	5.53616	5.51444	5.51284	5.51431	5.49982	5.50139
98	5.75198	5.71852	5.73632	5.71639	5.69125	5.71015
92	5.89954	5.84221	5.78511	5.87556	5.81237	5.75339
76	6.08668	6.05647	6.02628	6.04649	6.00584	5.99615
Stress (MPa)	4 Teeth Cosine 15mm	4 Teeth Cosine 20 mm	4 Teeth Cosine 25 mm	4 Teeth Tapered 15 mm	4 Teeth Tapered 20 mm	4 Teeth Tapered 25 mm
182	3.96993	3.89702	3.80882	3.76462	3.69706	3.65108
138	4.55524	4.53307	4.51613	4.49087	4.47150	4.45096
126	4.88990	4.99687	4.85663	4.76932	4.83272	4.63325
115	5.58857	5.58717	5.58426	5.55186	5.54083	5.52832
98	5.72146	5.73266	5.77464	5.71566	5.69664	5.66912
92	5.91849	5.89997	5.89917	5.88788	5.86889	5.82979
76	6.12004	6.10001	6.07900	6.09980	6.04927	6.02984

manufactured extrusion products were examined by using the same testing apparatus. When the obtained results were examined; it was observed that, fatigue cycles of damage occurrence are extremely close but still products manufactured by using dies with shorter land length have better fatigue strength. It is observed from the results that for 6 teeth gear products cosine curved die with 15 mm die land length is best die setup for higher fatigue resistance. Products manufactured by using dies with tapered cross have 3% less fatigue strength than comparing to dies with cosine curved profile. Cycle which the gear can be operated without getting any damage also decreases inversely proportionally with the increase of die land length.

When the fatigue strength diagrams which were obtained for parts manufactured with cosine and tapered dies by the four teeth forward extrusion methods were examined, a view similar to direct extrusion of six teeth parts can be observed. It was observed that for cosine and taper profiled dies' fatigue strength values are close to each other and by the increase of die land length, the fatigue strength values decrease too little. Detailed stress-cycle values for fatigue test results are given in Table 4.

Regardless of the fact that fatigue life cycle values are very close to each other in logarithmic scale, the measured cycles are quite different between each other. For example, when the fatigue stress value is 182 MPa, 6 teeth cosine curved die transition profile with 15 mm. die land length gives 3.36654 when for 20 mm. the die land length, cycle in logarithmic scale falls only 3.33746 but in the experiments, the difference between the cycle values is 150. The difference becomes bigger when the fatigue stress is lower. For the same die sets when the stress was chosen as 76 MPa, the difference between cycle values is obtained as 82040.

Comparison of experimental results with ANN

In order to verify the developed ANN models, a comparison between ANN predicted and experimental results was performed and presented in Fig. 7. The correlation coefficient (R^2) between predicted and experimental values of fatigue life for tapered die profile is 0.946 and for the cosine curved die profile, it is 0.999. The high values of R^2 indicates a fine agreement between the ANN-predicted and experimental values of fatigue life. It can also be seen in Fig. 8 that ANN predicted results are very close to experimental ones. The differences between the experimental and predicted values are reported as graphically in Fig. 9. Maximum percentage error in the test results of cosine curved die model is 2.95%, meanwhile maximum error of the tapered die model is 6.41%. Likewise, right y-axes of same figures show

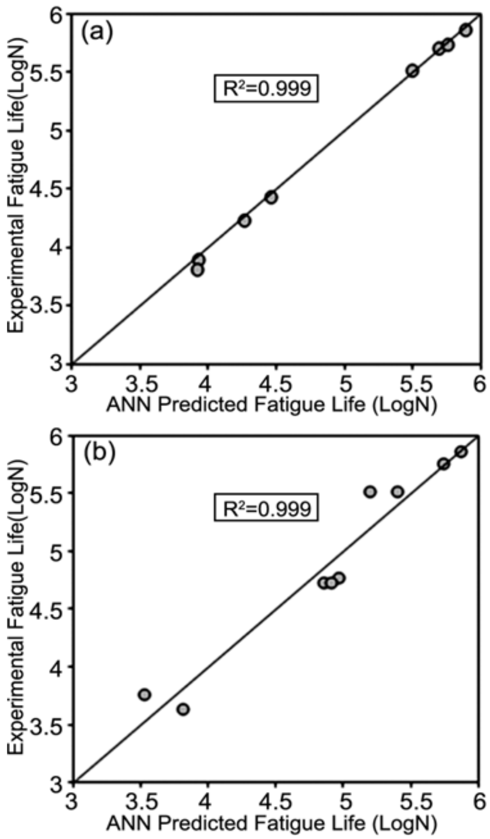

Fig. 7—Comparison of ANN-predicted and experimental values of fatigue life for (a) cosine curved and (b) straight tapered die profiles

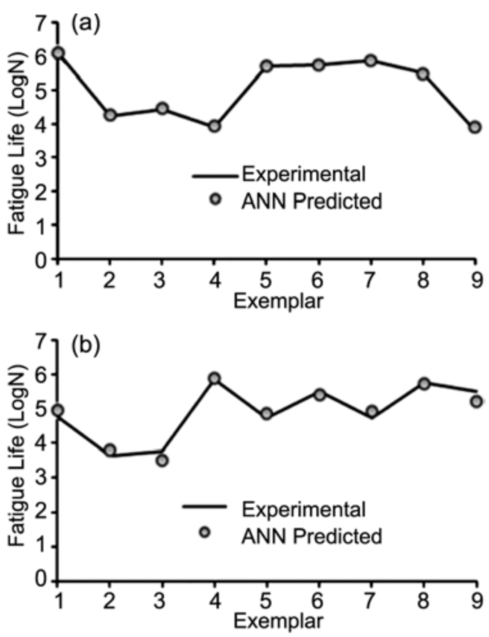

Fig. 8—Experimental and ANN predicted fatigue life results in the test period for (a) cosine curved and (b) straight tapered die profile

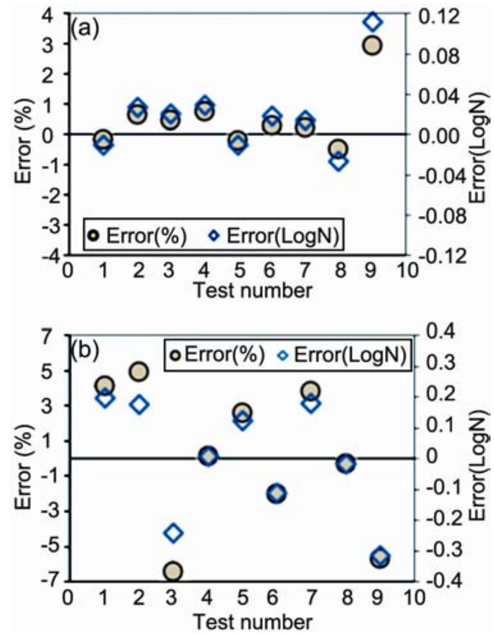

Fig. 9—Percentage error of the (a) cosine curved and (b) straight tapered die profile network models

the absolute error values. The maximum absolute error values for cosine curved model are in range of (-0.026)-(0.112) while the maximum absolute error values for tapered model are in range of (-0.241)-(0.199). These small error values indicate that predicted fatigue life results for both die profiles match well with the experimental results.

Conclusions

Fatigue life of an extruded product is affected by a number of critical parameters. Identifying the effects of process parameter plays a key role in the prevention of fatigue failure. The aim of the presented study is to investigate the fatigue strength of the gear like components. Additionally, fatigue strength of two different die profiles were also estimated by using ANN. The results obtained lead to the following conclusions:

(i) It was determined that extrusion products of 4 teeth parts' fatigue strength values vary according to die land length and transition profile and can be realized that die transition profile of 4 teeth dies produces more deformation than tapered dies and in this case the product's fatigue strength increases. Similarly to this observation, for 6 teeth products' fatigue cycles of damage occurrence are extremely close to 4 teeth samples but still products manufactured by using dies with shorter land length have better fatigue strength

(ii) It was also determined that for cosine and taper profile dies' fatigue strength values are close to each other and by the increase of die land length, the fatigue strength values decrease too little.

(iii) A good agreement between ANN-predicted and experimental fatigue strength results were obtained for different gear profiles. The R^2 between predicted and experimental data was 0.946 for straight tapered die profile and 0.999 for cosine curved profile.

(iv) The error values of the developed ANN models are 2.95% and 6.41% for cosine curved and straight tapered die profiles, respectively. Low error values indicate that complex fatigue

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