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INTEGRATED APPROACH TO AN OPTIMAL AUTOMOTIVE TIMING CHAIN SYSTEM DESIGN

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ABSTRACT−Ensuring engine efficiency is a crucial issue for automotive manufacturers. Several manufacturers focus on reducing the time taken to develop and introduce brand new vehicles to the market. Thus, a synergic approach including various simulations is generally adopted to achieve a development schedule and to reduce the cost of physical tests. This study involved proposing a design process that is very useful in the preliminary development stage through effective support from simulations. This type of simulation-based design process is effective in developing timing chain drives; the use of this process, based on results from multiple trials, showed improvements in performance including low friction and vibration, improved durability, and cost-effective part design when compared to conventional processes. This study proposes an integrated approach to the preliminary design of an automotive timing chain system. The approach involves structural and dynamic analyses. The details of the design process are described by using the case of a virtual engine. This study conducted and summarized a dynamic and structural analysis as well as topological optimization to describe a process to obtain optimal results. The results of this study indicated the following improvements in overall performance factors: 12.1 % improvement in transmission error, 10.1 % reduction in chain tension, 46 % reduction in tensioner arm weight, and 11 % reduction in transversal displacement.

KEY WORDS : Timing chain system, Design process, Dynamic analysis, Structural analysis, Topology optimization

1. INTRODUCTION

Chains are widely used in the modern automotive industry as timing drive systems that bind camshafts to crankshafts given the advantages of the timing chain system in terms of durability and engine size reduction. Conversely, the disadvantages of a chain timing system include vibration, noise, and frictional losses; thus, these factors should be carefully treated in the design process to reduce their negative influence on engine performance (Schoeffmann *et al*., 2015).

Typically, the performance of a chain system is generally influenced by chain tension and transmission errors. These factors affect engine performance, noise, and vibration generation. These problems are caused by the discrete nature of a chain that induces polygonal action and collisional contact between the chain and sprocket (Sopouch *et al*., 2002).

A timing system with a chain can be simplified as a set of chords that can be considered as multiple chain links. These links can cause vibrations in the transversal direction and longitudinal direction given the changes in the applied tension based on the chain operation that moves from the slack side to tight side (Morrison, 1952; Song *et al*., 2003). On the other hand, collisional contact occurs in the process of wrapping the sprocket with multiple linkages of the chain, and it can induce high frequency vibrations that result in noise at a high rpm of the engine (Sopouch *et al*., 2002).

Another problem involves the transmission errors that arise from crankshaft fluctuation. A modern automotive engine is generally operated by combustion; combustion causes a fluctuation in the rotational speed of the crankshaft owing to the inertial force of the piston. These fluctuations are transmitted to the chain, causing transmission errors in rotational motion and excessive wear of the chain. Therefore, the chain operation path is retained by several guides to reduce the effect due to fluctuations. Additionally, several commercial chain systems employ a pivot type guide with a hydraulic lash adjuster to rotate the pivot guide in accordance with the transversal reaction of the chain. Generally, this pivot guide is effective in reducing the vibration of the chain. Owing to the afore-mentioned guide system, tension is initially applied to the entire chain even if the engine is stationary (Sakaguchi *et al*., 2012).

As discussed above, several factors should be considered when designing a chain system; thus, the application of computational analysis in designing a timing chain system is an effective method to reduce development costs

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(Schoeffmann *et al*., 2015).

Traditional design rules for a timing chain system are expected to calculate various parameters including sprocket wrap angle and tensioner position. However, recently, multi-body dynamics are generally employed in the automotive industry to create a conceptual design for developing an accurate timing chain system. The computational method includes traditional design considerations. Furthermore, the method can supplement a polygonal action that affects fluctuations in crankshaft rotational velocity (Calabretta *et al*., 2011; Fuglede and Thomsen, 2016).

In this study, an integrated approach was proposed for the optimal design of an automotive timing chain system. The approach can be divided into structural and dynamic analyses. The details of the design process are described with the example of a case involving a virtual engine that imitates a commercial engine as closely as possible. In the study, a detailed process is elaborately described to obtain optimized results; the description also includes a comparison with the performance of an original chain system. Furthermore, this study focused on obtaining system parameters related to chain systems by analyzing results that involve parameters that are important in understanding chain behavior.

2. CASE STUDY

An integrated approach of the timing chain system design process begins with measurements involving a prototype engine test or the calculation of a mathematical model of the overall engine system. These basic data are important to realize reliable simulation models and to minimize the number of physical prototypes. Figure 1 shows the flowchart of the process for developing an optimal chain system from a prototype engine.

The process includes dynamic analysis, structural analysis, and topology optimization that are inter-related. For example, the boundary condition of structural analysis was obtained from a dynamic analysis that was based on prototype engine data. A result from structural analysis was employed to create a topology optimization model. The implemented result was applied to the dynamic analysis model to verify effectiveness.

2.1. Engine Hardware

Table 1 lists the specifications of the conceptualized engine used in this study. This engine closely imitated a commercial engine to the maximum possible extent, and the specification is assumed to be similar to that of an actual engine.

2.2. Geometrical Consideration

Figure 2 shows the timing drive side of a four cylinder DOHC gasoline engine that was considered as the base model in this study. The proposed model includes a machinable area to avoid interference while fixing components such as the fixed guides, tensioner, and tensioner arm. Furthermore, it was assumed that the available height of the chain system corresponded to 40 mm from the engine casing.

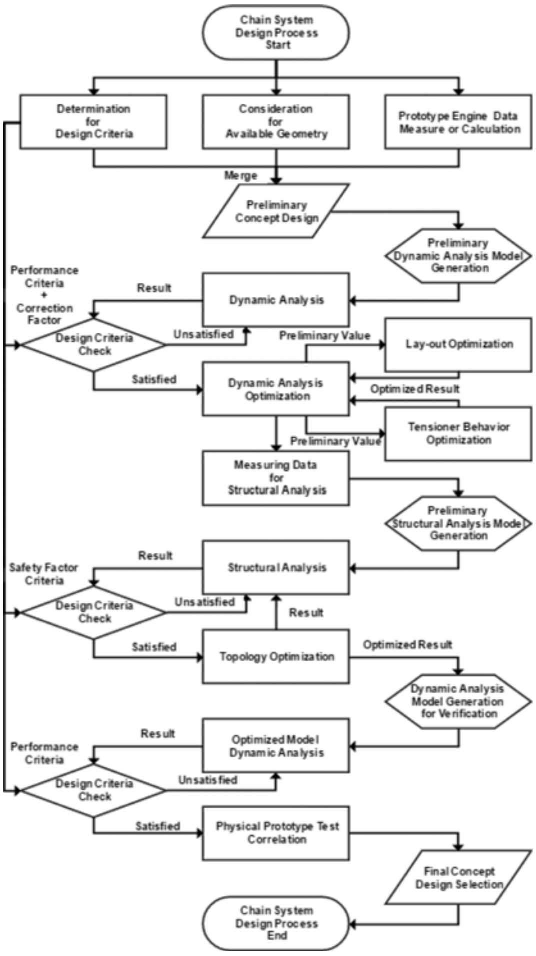

Figure 1. Integrated design process for the timing chain system.

Table 1. Specification of the conceptualized engine.

Engine specification	Unit	Value
Displacement	L	1.4
Cylinder number	cyl	4
Cylinder arrangement	-	Inline
Combustion	-	Gasoline
Number of valves	ea	16
Rated power	kW	75
Max. torque	Nm	130
Min. engine speed	rpm	850
Max. engine speed	rpm	6500

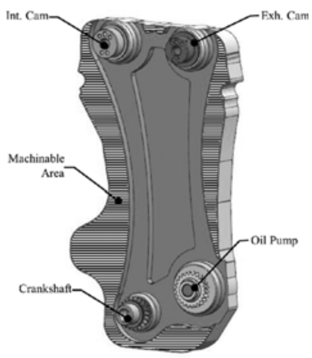

Figure 2. Geometry of the base model with a machinable area (The surface marked with a striped pattern).

Table 2. Sprocket specifications.

Part name	Q'ty	Property	Geometry
Crankshaft sprocket	1	Drive	Teeth: 21 and Width: 8.35 mm
Camshaft sprocket	2	Driven	Teeth: 42 and Width: 9.3 mm
Oil pump sprocket	1	Driven	Teeth: 23 and Width: 8.35 mm

In the timing chain side, it was assumed that two camshaft sprockets, an oil pump sprocket, and a crankshaft sprocket were included. Table 2 lists the specifications of each sprocket. The spline of the sprocket was potentially adopted from a commercial sprocket, and the width of the sprocket was selected in accordance with the applied torque. Therefore, the selected chain properties included a 6.35 mm long and an 8 mm wide pitch.

2.3. Inputs from the Engine Side

It was assumed that the base model was operated by the combustion of gasoline, which causes a fluctuation in the angular speed during the rotation of the crankshaft. These fluctuations were applied to the chain to cause several problems such as transmission errors on the rotational speed of the crankshaft (Priebsch *et al.*, 2004). Therefore, it was necessary to apply the measured fluctuation data to the dynamic analysis model to calculate chain tension and vibration. Additionally, driven sprockets, such as camshafts and oil pumps, absorb the transferred torque from the crankshaft. Thus, torque absorption should be added as the boundary condition for the model analysis.

Four different types of data were used in the dynamic analysis model for this study that included crankshaft fluctuation, camshaft torque and timing, and oil pump torque. Although the data were not derived from a real engine, they followed trends in amplitudes and frequencies that were similar to that of a real engine. The data were obtained in terms of specific engine speed, and the values between the speeds were interpolated on the basis of simulations.

2.4. Design Criteria

The main performance criteria of a chain system involve three different factors related to chain reliability, engine performance, and NVH. The chain reliability factors include maximum and minimum chain tension. Minimum chain tension is related to stable contact between a chain and sprocket, and maximum chain tension involves chain elongation and durability. The range was determined by the chain elongation test result, which generally allowed a 0.5 % longitudinal stretch. With respect to the case in this study, the range was determined from 10 N to 2.5 kN.

Factors related to engine performance included transmission errors. A transmission error can be defined as the chain that was ahead or behind the position in which the chain was originally placed. Chain system manufacturers set the error limit as within 6 degrees of the cam angle. Thus, the criterion in this study was set as ± 3 degrees of the cam angle.

The NVH issue was determined with the transversal displacement of the chain. Thus it was necessary to implement the following: placement of the measuring point in the middle of the free span and maintaining the displacement to below 5 % of the free span length. And for the guided section of the chain, the chain lift off from the guide should be lower than 1 mm. The above mentioned criterions are established from the generally accepted criterions by the component manufactuers and concerned ISO Standards (International Standards Office, 2004, 2005, 2006, 2015).

2.5. Conceptual Lay-out Selection

Geometry should be considered to avoid interference with existing parts at the very first stage of the layout design as the base model was assumed to have a restricted machinable area and height. Additionally, it was necessary to consider the design criteria involving performance factors. A cost aspect approach was also followed to optimize productivity.

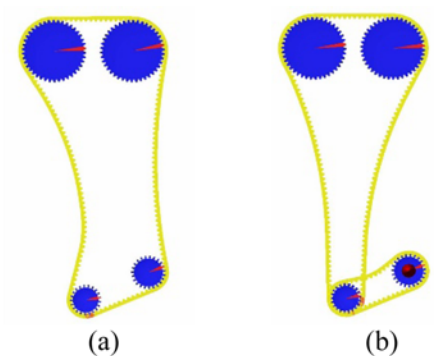

Figure 3. Conceptual layout for the base model: (a) One chain concept; (b) Separated oil pump drive concept.

Two different conceptual layouts were proposed based the geometry of the base model, and the layouts are shown in Figure 3. The separated drive conceptual model had an advantage in terms of durability as the chain transferred less power when compared to that of the one chain concept. However, relatively many components are required to operate the system, and this involves disadvantages in terms of quality control and costs. The separated chain case involves increased engine length due to the essential double-stage sprocket in the crankshaft position.

The one chain model has disadvantages in terms of acoustic behavior. However, this can be reduced by employing an inverted tooth chain and compatible sprockets. Therefore, in this study, the one chain conceptual model with an inverted tooth chain layout was selected as the base model.

3. DYNAMIC ANALYSIS

The model for analyzing multi-body dynamics was prepared with the extracted data as mentioned in the previous section. The model had a set of pivot guides and a tensioner at the slack side of the chain system and a fixed guide at the tight side. This constituted a basic approach to measure transversal displacement at the free span. Figure 5 shows the basic model and attached points of the main parts such as the tensioner. The commercial software, FEV Virtual Engine, has utilized for multi-body dynamics in this paper. The adopted software is an industry-specific vertical based on MSC Adams, which provide all templates necessary to create dynamics models of combustion engines and the powertrain. The model was generated with properties as specified in Table 3.

The boundary condition applied to the multi-body dynamics model has employed as simplified form which is based on the kinematic calculation results of the engine that has similar specification with the conceptualized engine mentioned in Table 1. The adopted inputs are single valve train (SVT) torques for intake and exhaust side, cam timings with reference to firing top dead center (FTDC) in cam angle, crankshaft speed fluctuations, and necessary torque to operate the oil pump. Figure 4 shows the input datas and the applied location is described on Figure 5.

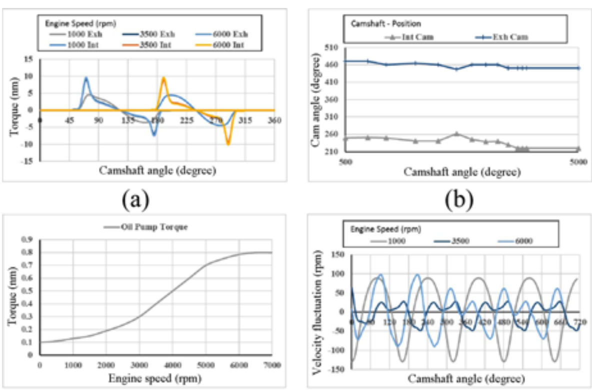

Figure 4. Input boundary condition of dynamic analysis model: (a) Cam torque; (b) Cam timing; (c) Oil pump torque; (d) Crank speed fluctuation.

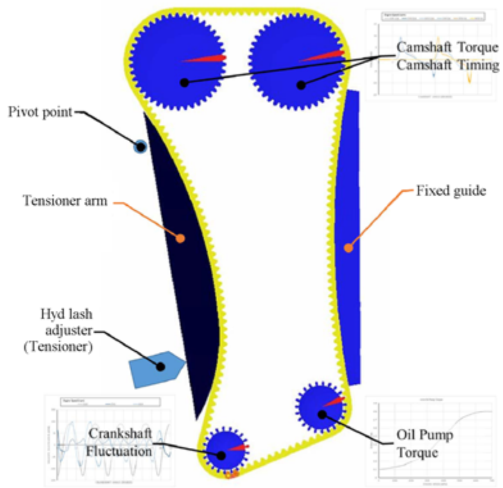

Figure 5. Base model for dynamic analysis and relevant inputs.

Table 3. Timing chain system properties.

Part name	Q'ty	Specification
Chain	162	Inverted tooth chain, Width 8 mm Pitch 6.35 mm, Pitch to back 3 mm
Tensioner arm	1	Material: Aluminum, Mass: 0.2 kg
Fixed guide	1	Material: Aluminum, Mass: 0.1 kg
Hydraulic lash adjuster	1	Check valve on housing, Relief valve

The pivot guide was designed for the base dynamic analysis model in the form of mass parts without any structural strengthening such as that offered by rib shapes. The considered fixed guide involved a plate with a curved surface as the actual parts were typically under endurable stress with respect to the aluminum structure. The final shape could be changed during optimization if an unexpectedly high stress was applied. The moment of inertia calculated from the mass model was applied to the dynamic analysis model and was used for gathering initial data for optimization.

The tensioner is referred to as a hydraulic lash adjuster and is described as shown in Figure 6. The effect of tensioner spring stiffness corresponds to a controllable factor that can affect the chain system performance in terms of transversal displacement of the chain (Takagishi *et al*., 2008).

The analysis was performed with an rpm sweep condition under a full load. The analysis commenced at

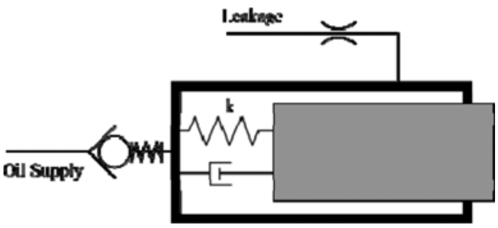

Figure 6. Model of hydraulic lash adjuster (Tensioner).

500 rpm and was completed at 7000 rpm. This range has an additional margin from the rated minimum and maximum engine speed range from 850 rpm to 6500 rpm. However, data were gathered only from the engine speed range. The simulation condition was set as 144000 steps for 6.4 s for the mentioned rpm sweep.

3.1. Lay-out Optimization

The layout of the timing chain system affected the initial tension that was strongly related to the performance of the timing system. It could be rearranged by modifying the curves of guides and pivot points of the tensioner arm. The initial tension of the system had an obvious optimal point that minimized chain tension and transmission error. Several models with different initial tensions were tested to find the optimal value. Figure 7 describes the test results.

Figure 7 shows the optimal initial tension for the base model to be 85 N, and thus this tension was employed in further steps. The chain length to induce the afore mentioned initial tension corresponded to 1026.7 mm, including an acceptable 0.4 % elongation that exceeds the original length of 1022.3 mm. The optimized initial tension resulted in maximum average reductions of 12.1 % and 10.1 % in transmission error and chain tension, respectively.

It is also necessary to address transversal displacement for optimization. There were two free spans on the base model such as the space between the exhaust cam and intake cam and that between the oil pump sprocket and crankshaft. As previously mentioned, the transversal displacement should be below 5 % of the free span length at the middle of the free span. The distance between cams corresponded to 80 mm, and thus the displacement should lower than 4 mm. The crankshaft side distance is approximately 60 mm, which has a displacement limit of 3 mm.

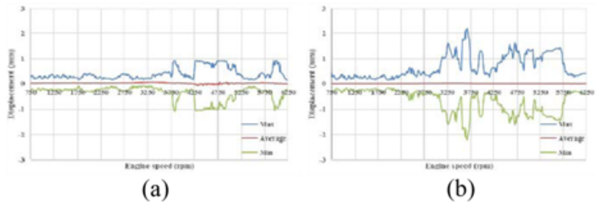

Figure 8. Transversal displacement of base model: (a) Camshaft side; (b) Crankshaft side.

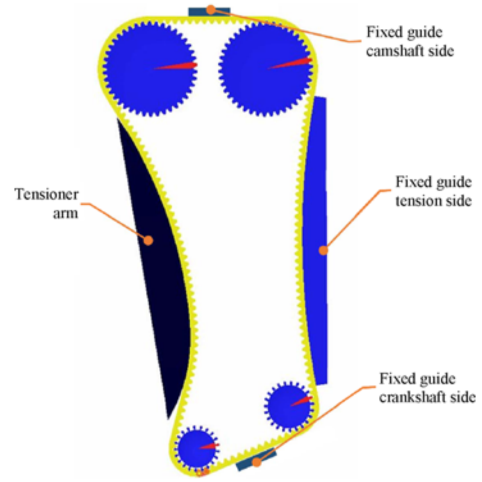

Figure 9. Optimized layout with four guides and 85 N of initial tension.

Figure 8 shows the measured displacement data from the base model, and the maximum displacement was calculated as approximately 1.82 mm for the camshaft side and 4.38 mm for the crankshaft side. The measured values corresponded to a low margin with respect to the displacement limit. Hence, an additional fixed guide for the free span was added to the model with a gap of 1 mm from the backside of the chain.

The final layout in the optimized model has three fixed guides, a tensioner arm, a hydraulic lash adjuster, four sprockets for the inverted tooth chain, and 162 links of the inverted tooth chain with a pitch of 6.35 mm of. Figure 9 shows the layout.

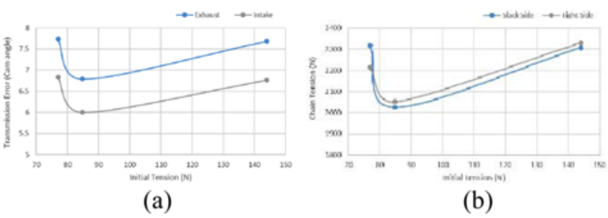

Figure 7. Performance of the timing system based on initial tension: (a) Transmission error; (b) Chain tension at slack and tight side.

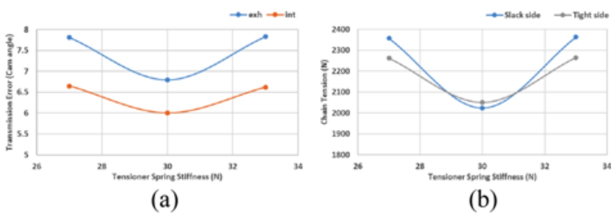

Figure 10. Timing system performance based on tensioner spring stiffness: (a) Transmission error; (b) Chain tension at slack and tight sides.

The behavior of the tensioner was optimized by changing the spring stiffness by ± 10 % from the standard value of 30 N. Figure 10 describes the analysis results, and the graph shows that the optimal value corresponds to approximately 30 N. The optimized stiffness of the tensioner spring allowed maximum reductions of 13.9 % in transmission error and 11.5 % in chain tension, respectively.

3.2. Analysis Results

Figures 11 ~ 13 describe the dynamic analysis results of the optimized layout. The results reveal a critical frequency at 3,250 rpm, and various performance factors, such as chain tension, transmission error, and transversal displacement, indicate maximum values at the frequency.

The crankshaft speed at 3,250 rpm shows the larger variation in amplitude and magnitude of fluctuation than the applied speed plots at low engine rpm. The given situation worsens the engine performance including increase of the chain tension and transmission error. The data on Figures 11 ~ 13 also can be construed that the performance related parameters such as chain tension become worse while approaching to 3,250 rpm with instable crankshaft speed fluctuation, however, the tensioner works for compensating the drastic movement of the tensioner arm, then the performance parameters are changed into the stable condition.

The optimized layout exhibited an improvement of 12.1 % with respect to transmission error and a 10.1 % reduction in chain tension when compared to the base model.

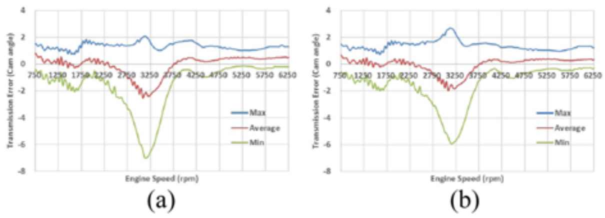

Figure 11. Transmission error of the optimized model: (a) Exhaust camshaft; (b) Intake camshaft.

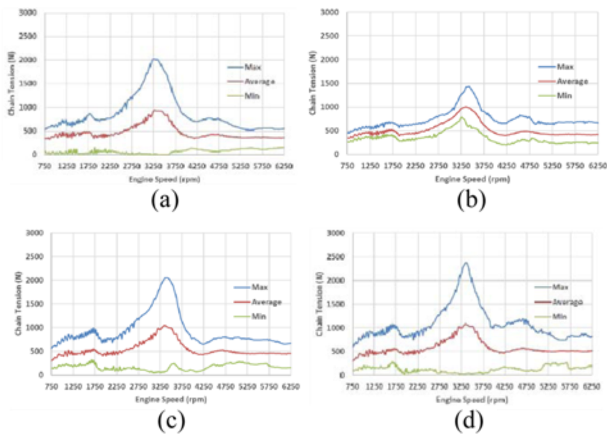

Figure 12. Chain tension of the optimized model: (a) Slack side; (b) Camshaft side; (c) Tight side; (d) Crankshaft side.

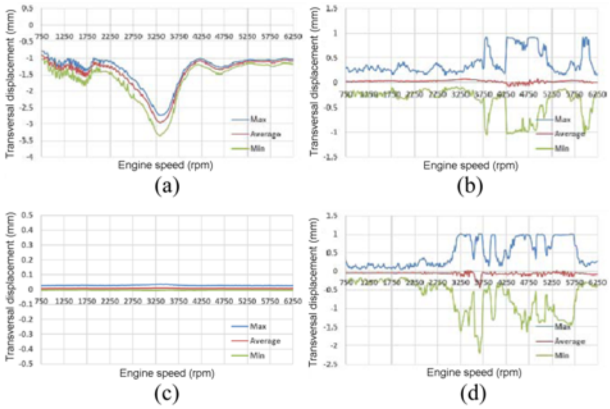

Figure 13. Transversal displacement of the optimized model: (a) Slack side; (b) Camshaft side; (c) Tight side; (d) Crankshaft side.

4. STRUCTURAL ANALYSIS

Structural analysis is necessary to calculate stress during system operation and to determine the necessity of structural reinforcements based on the calculation of the safety factor. The analysis in this study was in accordance with the results of the dynamic analysis data. As mentioned in the previous section, the fixed guide that was under endurable stress was considered for the aluminum structure, and thus the present study only considered the tensioner arm. The concerned analysis has performed by the commercial software common in the industry, Abaqus/Standard.

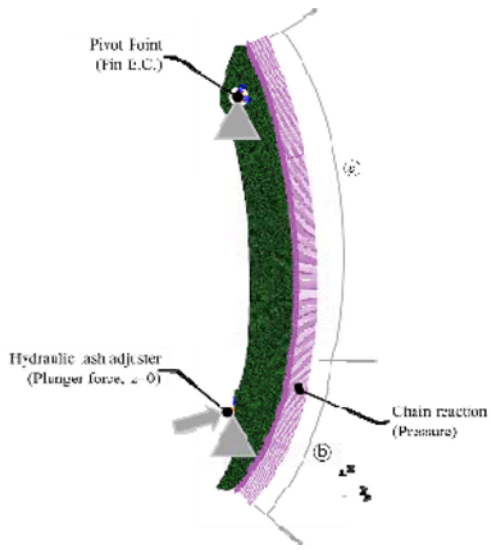

Figure 14. Applied boundary condition for the tensioner arm.

In the dynamic analysis model, it was assumed that the guides were formed with aluminum casting. Thus, the material properties of aluminum for structural analysis were as follows: density of 2.97 g/cc, elastic modulus of 71.0 GPa, and Poisson's ratio of 0.36 (Cubberly *et al*., 1979). The plastic deformation and relevant material properties were not considered on this analysis, since the system should not be deformed to maintain the performance related factors of the system such as the transmission error and chain tension.

Figure 14 shows the boundary condition applied for structural analysis. The pivot point was constrained with respect to the pin boundary condition. The plunger contact area had constraints involving a z-axis of zero, and the tensioner force was applied by a rigid body plunger. The tensioner force and chain reaction were extracted from dynamic analysis results, and the analysis result was correlated with the behavior of the tensioner.

4.1. Data Extraction from Dynamic Analysis

Applied boundary conditions for the structural analysis were extracted from the dynamic analysis as shown in Figure 15. The maximum force from the plunger was approximately 1 kN, and a radial directional force of 22.5 kN was applied by the chain at the central point of the tensioner arm at the same rpm.

The dynamic analysis indicated two different forces. Approximately 22.5 kN of the force was applied to 27 links from the camshaft (that is, ⓐ section of Figure 14), and 28.7 kN of the force was applied to 9 links from the crankshaft (that is, ⓑ section of Figure 14) as the lower side of tensioner arm was directly exposed to the tensioner plunger force. The pressure applied on each section was calculated from the given force, number of links, pitch of the chain, and chain width. Therefore, 410 kPa of pressure was applied to section ⓐ, and 510 kPa was applied to section ⓑ.

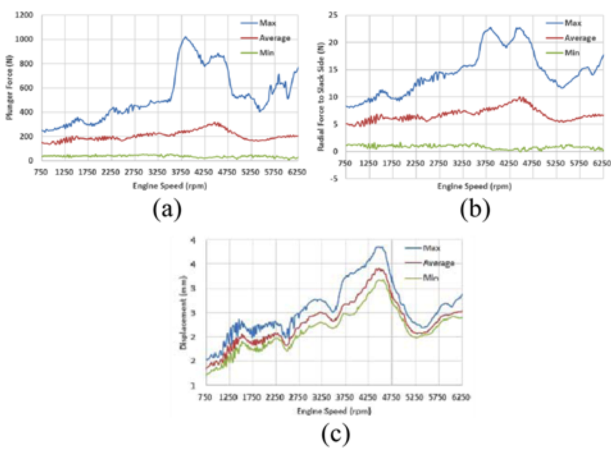

Figure 15. Extracted data from the dynamic analysis model: (a) Plunger force; (b) Radial directional force by the chain to the tensioner arm; (c) Tensioner plunger behavior.

The tensioner plunger behavior was extracted for the correlation with the structural analysis results. The simulation result indicates the displacement at the adjuster-attached position; this is similar to the plunger behavior that measured 2.8 mm at 3,750 rpm at which the maximum plunger force was applied.

4.2. Analysis Result

Figure 16 shows the results of thenalysis. The displacement at the plunger contact point was approximately 3 mm, which corresponded to 90 % of the dynamic analysis result. Therefore, the model considered was well as correlated with the dynamic analysis model. The results indicated several stress concentration points, such as inner and outer curves, caused by the bending moment. This type of uneven stress distribution was optimized by topology optimization.

The maximum stress that occurred at the plunger contact point corresponded to 23 MPa. This value was lower than 10 % of the ultimate strength of aluminum, and thus the structure could be optimized with a sufficiently high safety factor.

5. TOPOLOGY OPTIMIZATION

Topology optimization to reduce the weight of the tensioner arm was carried out in accordance with the structural analysis results. The commercial software common in the industry, Abaqus/ATOM, has utilized to carry out the analysis. The topology optimization in used software produces a new geometry by scaling the relative densities of the elements in the design domain. Elements with large relative densities are retained while the elements with sufficiently small relative densities are assumed to be voids. Condition based optimization, which uses the strain energy and the stresses at the nodes as input data and does not need to calculate the local stiffness of the design variables, was performed for the mass model (Abaqus, 2011).

In aspect of weight reduction by the topology optimization, the moment of inertia should be considered

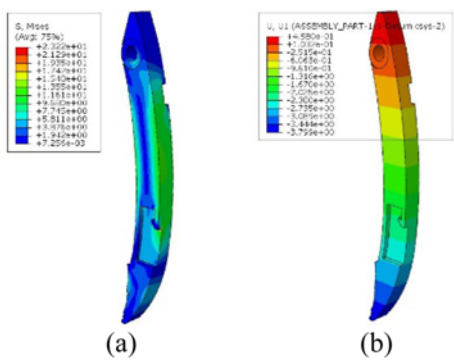

Figure 16. Tensioner arm structural analysis results: (a) Stress; (b) Displacement.

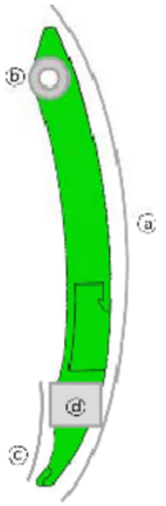

Figure 17. Geometrical restrictions for topology optimization.

since that the inertia is highly related with the critical frequency of the entire system which is affected by the tensioner arm. Also, considerable decrease of the mass inertia can induce the instability of the chain system. Therefore, the objective function in this paper was set as strain energy with a 30 % volume constraint. And according to the variation of the moment of inertia, the optimal mass-reducted shape has selected for designing final model of tensioner arm. Geometrical restrictions were applied to the contact areas and fixing parts as described in Figure 17. Specifically, ⓐ section corresponded to a chain contact point, ⓑ and ⓒ sections corresponded to a constraint for connecting parts, and ⓓ section was restricted with respect to the absorbing force from the plunger, which exhibited high stress as indicated by the structural analysis results.

5.1. Optimization Results

Figure 18 shows the topology optimization result, and Table 4 describes the moment of inertia variation by volume reduction which has resulted by topology optimization. The moment of inertia shows tendency with gradual decrase in accordance with the volume reduction. However, the optimized shape over the 43.5 % of volume reduction lost its structures to achieve necessary stability as shown in Figure 18 (c).

Table 4. MOI variation by volume reduction.

Volume reduction	I_{xx} (kg·mm^2)	I_{yy} (kg·mm^2)	I_{zz} (kg·mm^2)
0 %	2.09	0.98	1.11
25.5 %	1.58	0.78	0.81
29.9 %	1.48	0.74	0.74
34.6 %	1.38	0.69	0.68
38.0 %	1.28	0.66	0.62
43.5 %	1.17	0.61	0.55
51.6 %	1.00	0.54	0.46

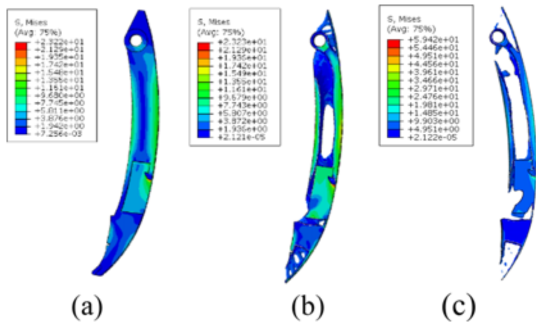

Figure 18. Topology optimization result: (a) Mass model shape; (b) Optimized model shape at 43.5 % of volume reduction; (c) Optimized model shape at 51.6 % of volume reduction.

The previously mentioned condition based optimization involved 15 calculated serial steps, and this resulted in the shape as shown in Figure 18 (b) at 43.5 % of volume reduction. The shape described in this figure was considered as a reasonable optimization that showed a similar trend to that of the structural analysis result.

The tensioner arm was designed in accordance with the topology-optimized shape, and the new design is described in Figure 19. The new design shows a relatively well-distributed stress contour when compared to the mass model. However, the maximum stress was increased to 32 MPa. This value was still acceptable as the value was still lower than 15 % of the ultimate strength of aluminum.

Weight reduction was achieved by design optimization. The original mass model corresponded to 0.26 kg, and the optimized model had a weight of 0.14 kg, which

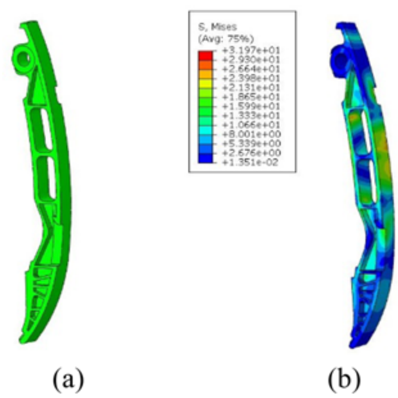

Figure 19. Optimized design based on topology optimization: (a) Designed shape; (b) Structural analysis result.

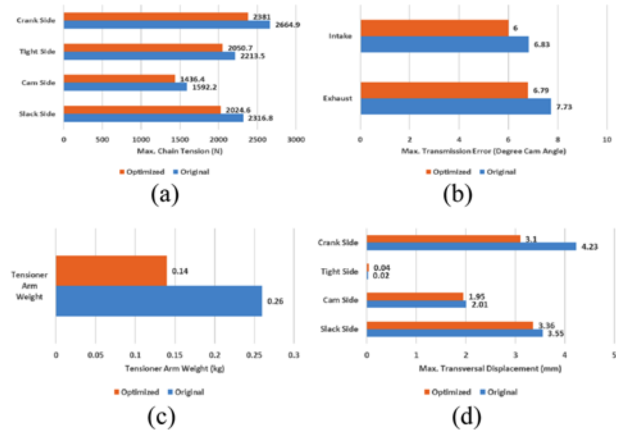

Figure 20. Performance factor comparison between base model and optimized model: (a) Maximum chain tension; (b) Maximum transmission error; (c) Tensioner arm weight; (d) Maximum transversal displacement.

corresponded to a reduction of 46 % when compared to the weight of the mass model.

6. CONCLUSION

An optimal design is extremely important in the modern automotive industry; it is suitable for designing a timing chain system given that the system involves several analytical difficulties that stem from polygonal action. In this study, an integrated approach was proposed for the optimal design of an automotive timing chain system. The details of the design process were described using the example of a virtual engine.

The process including dynamic analysis, structural analysis, and topology optimization was fully followed to obtain optimized results. This included a comparison with the performance of the original chain system, as shown in Figure 20. The result showed an improvement with respect to overall performance factors, a 12.1 % improvement in transmission error, a 10.1 % reduction chain tension, and a 46 % reduction in tensioner arm weight. The maximum transversal displacement (that is, vibration) was reduced with respect to the overall position with the exception of the tight side. However, the measured value was still acceptable with respect to the criterion wherein the lift off from the guide should be lower than 1 mm.

The process described in this study saves expenses in terms of physical engine testing and can be easily adapted for manufacturers who are likely to start a timing chain manufacturing business.

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