Design and Vibration analysis of the crank Shaft of the power loom system in the textile industry

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ABSTRACT- Crank shaft of Power loom system in textile machinery is a critical component which transmits the power from motor to machinery at a required speed and controls the complete operation. If crank shaft of the system fails then entire process will stop hence it is important to design the power loom system for continues operation. A crank shaft of length 1806 mm length is to be redesigned to transmit the power for a given conditions. This work focus on calculation of physical properties such as stress, deflection and modal frequencies for the existing component using the FEA package and modal frequencies are further validated by experimental results using FFT analyser. Existing component is to be redesigned and remodelled using modelling software for different dimensions of crank web and diameter of the shaft then static structural and dynamic analysis of modelled components is to be carried out using the FEA package for safer work.

Key words: Crankshaft, crank pin, FEA, FFT analyzer, bending moment, von mises stresses, deflection

1. Introduction

The textile industry plays very important role in Indian economy. It ranks second after the agriculture. Besides the industrial production, it provides employment to millions of the people of the country and gives a handsome earning of foreign exchange through export. It is also source of livelihood in villages and remote areas. Millions of people in our country depend upon it. The rapid growth and development of this industry is remarkable.

The Indian textile industry is consisted of the following three groups

i) Mill Sector
ii) Handloom Sector
iii) Power loom Sector

The power looms are one of the most important equipment in producing of cotton terry towels and bed sheets. The power looms are used for weaving the dyed yarn to towels and bed sheets

The basic purpose of any loom is to hold the warp threads under tension to facilitate the interweaving of the weft threads. A loom is a tool used for weaving yarn into textiles.

A power loom, yet another type of loom, is a mechanized tool that uses a drive shaft for power. Invented by Edmund Cartwright in Great Britain in 1784, the power loom allowed manufacturers to create textiles much more quickly than with hand-driven looms. This improvement helped the power loom become one of the defining machines of the industrial revolution. A loom works by holding lengthwise threads, called the warp, under tension. The vertically-oriented threads are attached to two or more harnesses which move up and down, separating warp threads from each other and creating a space called the shed. Another thread, called the weft, is wound onto spools called bobbins, which are placed in a shuttle and passed through the shed, which creates the weave. In the early 20th century, the shuttle less loom, also known as the rapier loom, was invented. This type of power loom moves the weft through the shed using jets of air or water, steel rods, or a dummy shuttle that leaves a trail of yarn rather than using a weft

A crankshaft is a mechanical part able to perform a conversion between reciprocating motion and rotational motion. In order to do the conversion between two motions, the crankshaft has "crank throws" or "crankpins", additional bearing surfaces whose axis is offset from that of the crank. Crank shaft of Power loom system in textile machinery is a critical component which transmits the power from motor to machinery at a required speed and controls the complete operation. If crank shaft of the system fails then entire process will stop hence it is important to design the crank shaft of power loom system for continues operation.

2. Literature Review

Shin-Yong Chen, Chieh Kung, Jung-Chun Hsu et al [1] in their research paper “Dynamic Analysis Of a Rotating Composite Shaft” One of the key factors in designing a motor built-in high speed spindle is to assemble the motor rotor and shaft by means of hot-fit. Presented in this paper is a study of the influence of a hot-fit rotor on the local stiffness of the hollow shaft. Dynamic analyses of the rotor-hollow shaft assembly using contact elements are conducted. The normal contact stress state between the rotor and the hollow shaft is obtained through the use of contact elements with friction effects included. The normal contact stress, considered as the pr-stress between the rotor and the hollow shaft, is then adopted for subsequent modal analyses. In this study, the modal analysis results are verified by a modal testing experiment

Neepa M. Patel et al [2] have analyzed that to increase the productivity of clothes without selvage, shuttle loom is necessary, which produce clothes at lower cost. The only drawback of shuttle loom is its low speed, current shuttle looms are running at 120 ppm(pick per minute), and due to this its productivity is less. Therefore, in this paper kinematic and dynamic analysis has been done for present and proposed mechanism, to design high speed Beat-up mechanism, which is 3rd primary operation of shuttle loom. Basically beat-up mechanism is the reciprocating motion of the reed which is used to push every weft thread to the fabric fell

C. AZOURY et al [2] presents a report on the experimental and analytical modal analysis of a crankshaft. The effective material and geometrical properties are measured, and the dynamic behavior is investigated through impact testing. The three-dimensional finite element models are constructed and an analytical modal analysis is then performed to generate natural frequencies and mode shapes in the three-orthogonal directions. The finite element model agrees well with the experimental tests and can serve as a baseline model of the crankshaft.

Ms.Shweta Ambadas Naik[4] carried out a review study on failure analysis of a crank shaft. In this paper, the stress analysis and modal analysis of a 4-cylinder crankshaft are discussed using finite element method. The review of existing literature on crankshaft design and optimization is presented. The materials, manufacturing process, failure analysis, design consideration of the crankshaft are reviewed here

R. J. Deshbhratar, and Y.R Suple et al [3] have analyzed 4- cylinder crankshaft and model of the crankshaft were created by Pro/E Software and then imported to ANSYS software The maximum deformation appears at the centre of crankshaft surface. The maximum stress appears at the fillets between the crankshaft journal and crank cheeks, and near the central point. The edge of main journal is high stress area. The crankshaft deformation was mainly bending deformation under the lower frequency. And the maximum deformation was located at the link between main bearing journal and crankpin and crank cheeks. So this area prone to appear the bending fatigue crack.

Abhishek choubey, and Jamin Brahmbhatt et al [4] have analyzed crankshaft model and 3-dimentional model of the crankshaft were created by SOLID WORKS Software and imported to ANSYS software. The crankshaft maximum deformation appears at the centre of crankpin neck surface. The maximum stress appears at the fillets between the crankshaft journals and crank cheeks and near the central point journal. The edge of main journal is high stress area.

Rinkle garg and Sunil Baghl et al. [5] have analyzed crankshaft model and crank throw were created by using Pro/E Software and then imported to ANSYS software. The result shows that the improvement in the strength of the crankshaft as the maximum limits of stress, total deformation, and the strain is reduced. The weight of the crankshaft is reduced .There by, reduces the inertia force. As the weight of the crankshaft is decreased this will decrease the cost of the crankshaft and increase the I.C engine performance.

Sanjay B Chikalthankar et al [6] investigated stresses developed in crankshaft under dynamic loading. In this study a dynamic simulation was conducted on crankshaft, Finite element analysis was performed to obtain the variation of stress magnitude at critical locations. The pressure-volume diagram was used to calculate the load boundary condition in dynamic simulation model, and other simulation inputs were taken from the engine specification chart. This load was then applied to the FE model, and boundary conditions were applied according to the engine mounting conditions. The analysis was done for different engine speeds and as a result we get critical engine speed and critical region on the crankshaft. Stress variation over the engine cycle and the effect of torsional load in the analysis were investigated. Results obtained from the analysis are very useful in optimization of this crankshaft.

Sagar R Dharmadhikari, et al [7] made modest attempt to review the optimization of Genetic Algorithm and ANSYS in their research report "Design and Analysis of Composite Drive Shaft using ANSYS and Genetic Algorithm". Drive shaft is the main component of drive system of an automobile. Conventional steel is substituted by composite material which has high specific strength and stiffness. The finite element analysis is used in this work to predict the deformation of shaft. Natural frequency using Bernoulli – Euler and Timoshenko beam theories was compared. The frequency calculated by Bernoulli – Euler theory is high because it neglects the effect of rotary inertia & transverse shear. Hence the single piece High Strength Carbon / Epoxy composite drive shaft has been proposed to design to replace the two piece conventional steel drive shaft of an automobile.

K. Thriveni Dr.B.JayaChandraiah et al [8] made an attempt in this paper to study the Static analysis on a crankshaft from a single cylinder 4-stroke I.C Engine. The model of the crankshaft is created using CATIA-V5 Software. Finite element analysis (FEA) is performed to obtain the variation of stress at critical locations of the crank shaft using the ANSYS software and applying the boundary conditions. Then the results are drawn Von-misses stress induced in the crankshaft is 15.83Mpa and shear stress is induced in the crankshaft is 8.271Mpa. The Theoretical results are obtained von-misses stress is 19.6Mpa, shear stress is 9.28Mpa. The validation of model is compared with the Theoretical and FEA results of Von-misses stress and shear stress are within the limits. Further it can be extended for the different materials and dynamic analysis, optimization of crank shaft.
Ashwani Kumar Singh et al [9] conducted statics analysis on a nickel chrome steel and structural steel crank shafts from a single cylinder four stroke engine. Finite elements analysis was performed to obtain the variation of stress magnitude at critical locations.

Three dimensional model of crankshaft was created in Pro/E software .The load was then applied to the FE model and boundary condition where applied as per the mounting conditions of the engine in the ANSYS Workbench

Abhishek Sharma et al [10] in the present research work vibration analyses have been focused to detect crankshaft fault at the early stage, followed by the literature review of the shaft and the experimental methodologies. A simulation for the study of crankshaft is carried out by acquiring its fault signal and its fast Fourier transform is plotted to show the characteristics frequencies and its harmonics. A comparison of simulated data is also made to validate the experiment based condition monitoring.

Momin Muhammad Zia Muhammad Idris et al [11] this paper presents results of strength analysis done on crankshaft of a single cylinder two stroke petrol engine, using PRO/E and ANSYS software. The three dimensional model of crankshaft was developed in PRO/E and imported to ANSYS for strength analysis. This work includes, in analysis, torsion stress which is generally ignored. A calculation method is used to validate the model. The paper also proposes a design modification in the crankshaft to reduce its mass. The analysis of modified design is also done

3.Problem identification

Crank shaft of Power loom system in textile machinery is a critical component which transmits the power from motor to machinery at a required speed and controls the complete operation. If crank shaft of the system fails then entire process will stop hence it is important to design the power loom system for continues operation. There is requirement from industry Dhayafule Textiles for the vibration analysis of the crank shaft of a power loom

4. Objectives

This work comprises the following objectives for safe design of existing power loom system.

- To select critical component i.e. crank shaft of power loom system from textile industry.
- To model the existing crank shaft of power loom system then numerical analysis will be carried out in FEA
- To experimentally analyze the existing component using FFT analyzer
- To compare the results and remodel the component
- If the component undergoes failure then the component is redesigned, static structural and modal analysis will be carried out

5. Failed Component

Figure 1 Failed component

6. Experimental Setup

The schematic diagram of the complete experimental setup is shown in following

Figure 2 Experimental set up

7. Experimental results

The natural frequencies obtained from the experimentation are listed below in table 1

Table1

Sl No	Mode	Modal Frequency in Hz
1	1	65.918
2	2	70.1532
3	3	71.125
4	4	78.776
5	5	338.101
6	6	342.843
7	7	350.359
8	8	366.877
9	9	615.127
10	10	658.241
11	11	694.890

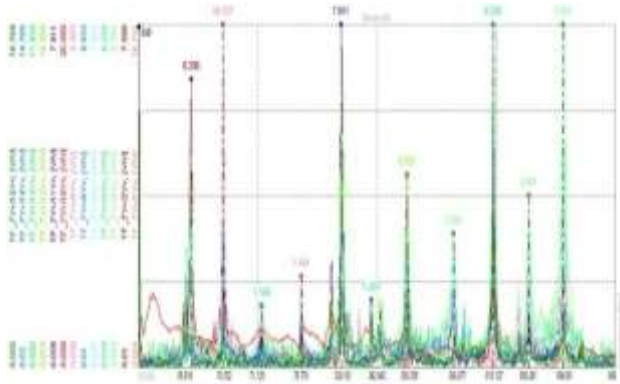

Figure 3 Graph of experimental results

8. Design Procedure for Crankshaft

The following procedure may be adopted for designing a crankshaft.
1. First of all, find the magnitude of the various loads on the crankshaft.
2. Determine the distances between the supports and their position with respect to the loads.
3. For the sake of simplicity and also for safety, the shaft is considered to be supported at the centres of the bearings and all the forces and reactions to be acting at these points. The distances between the supports depend on the length of the bearings, which in turn depend on the diameter of the shaft because of the allowable bearing pressures.
4. Now calculate the distances between the supports.
5. Assuming the allowable bending and shear stresses, determine the main dimensions of the crankshaft.

Geometric properties, Material properties of crank shaft

Table 2

Material	Cold drawn Steel 1018
Length	1806mm
Diameter	40 mm
Modulus of elasticity	210Gpa
Density	7850
Poisson's ratio	0.3

Operating frequency of the crank shaft

$f = \frac{\omega}{2\pi}$

N=120 rpm

$\omega = \frac{2\pi N}{60}$

f =2 Hz

8.1 Force acting on the crank shaft

A) Weight of slay = 600 N

B) Downward force acting by pulley

$\frac{p_1}{p_2} = e^{\mu\theta}$

$M_t = (p_1 - p_2)R_1$

$M_t = \frac{60x10^6(kW)}{2\pi n}$

$\frac{p_1}{p_2} = e^{0.24\pi} = 2.125$

$p_1 = 2.125p_2$

$M_t = \frac{60x10^6 x\ 0.746}{2\pi\ x\ 120} = 59364.79$ N-mm

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$(2.125p_2 - p_2)228.6$ = 59364.79

p_2= 490.5201N p_1= 230.883 N

W_1= mg =10.8x 9.8=105.84 N

Therefore the total downward force is given by

(p_1+p_2+W) =(490.520+230.883+278.3)= 1000 N

<u>C) Force acting by a gear</u>

$p_t = \frac{2M_t}{d_1}$

$p_r = p_t \tan \propto$

p_t= 492.034N

p_r= 178.5 N

<u>D) Force acting by a Sprocket</u>

Speed N = 120 rpm

Number of teeth = 12

$V = \frac{z_1 PN}{60X10^3}$ = 1.29 m/sec

P= $\frac{N\,X\,power}{V}$ = 75 N

8.2 Layout of the crankshaft

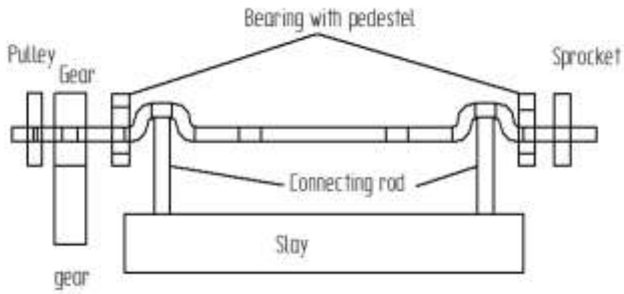


Figure 4 Layout of the crank shaft

Reaction Forces

Let the reaction at C and F be V_c and V_f respectively, taking moments about C we have,

61200+737400-1331V_f-111075-1077000-22312.5-225000=0

440212.5=1331V_f

V_f =330.73N

330.73+V_c+75 =2378.5N

V_c=1972.77 N

8.3 Bending moment of vertically acting forces

Bending moment at A, 0

Bending moment at B, 1,00,000 N-mm

Bending moment at C, 2,47,312.5 N-mm

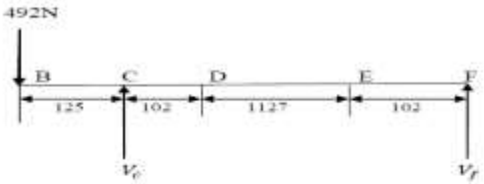
Bending moment at D, 166296.96 N-mm

Bending moment at E, -52645.33 N-mm

Bending moment at F, -11250 N-mm

Bending moment at G, 0

8.4 Bending moment of horizontally acting forces



Let the reaction at C and F be V_c and V_f respectively, taking moments about C we have,

1331V_f= -61500

V_f =-46.205N

V_c = 538.205N

Bending moment at B, 0

Bending moment at C, 61500 N-mm

8.5 Resulting bending moment at

A=0

B=$\sqrt{0+100000^2}$=1,00,000 N

C=$\sqrt{24731.5^2+61500^2}$=254844.50 N-mm

D=$\sqrt{166296.96^2}$=166296.96 N-mm

E=$\sqrt{(-52645.33)^2}$=52645.33 N-mm

F=$\sqrt{(-11250)^2}$=11250 N-mm

The material of the shaft is cold drawn Carbon steel 1018

S_{yt}=345MPa

S_{ut}=414MPa

Permissible shear stress

0.30S_{yt}=0.3(345)=103.5 N/mm^2

0.18S_{ut}=0.18(414)=74.52 N/mm^2

The lower of the two values is 74.52 N/mm^2 and there are keyways on the shaft

τ_{max}=0.75(74.52)=55.89 N/mm^2

$d^3 = \frac{16\sqrt{(k_b M_b)^2+(k_t M_t)^2}}{\Pi X \tau_{max}}$

$d^3 = \frac{16\sqrt{(2.5X254844.50)^2+(2.5X59364.79)^2}}{\Pi X 55.89}$

d=38.80 mm

The diameter of the shaft is obtained by considering the loads acting on the shaft

8.9 Design of Crank pin

$(M_b)_c = \left[\frac{\Pi x d^3}{32}\right]\sigma_b$

σ_b=75 N/mm^2(allowable bending stress for crank pin)

d=40mm

$(M_b)_c=(R_1)_v b_1$

$(M_b)_c$=1972.77X102

=201222.54

$\sigma_b = \frac{201222.5432}{\Pi x d^3}$

σ_b = 32.025 N/mm^2

The bending stress obtained is less than the allowable bending stress for the crank pin hence the diameter assumed for the crankpin is safe.

9 Boundary conditions

- All degree of freedom of the Component at a distance of 226 mm from left side and 151 mm from right hand side end were arrested (Bearing locations)
- A load of 1000N is acting at a distance of 50mm from right side (Pulley location)
- A load of 492 N (tangential) and 178.5 N (downward) acting at a distance of 126 mm from right hand side (Gear location).
- A load of 75N (upward) acting at a distance of 51 mm from left hand side (sprocket location).

- A load of 600 N (downward) acting at middle of crankpin as shown in figure.

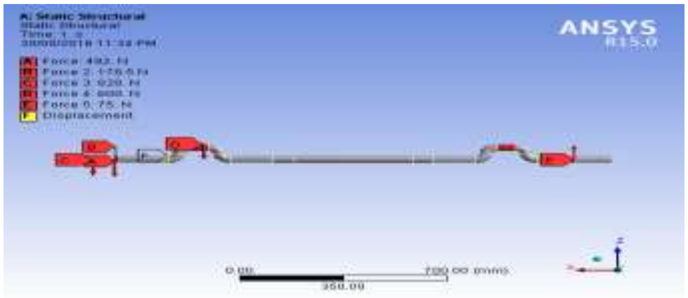


Figure 5 Boundary Condition

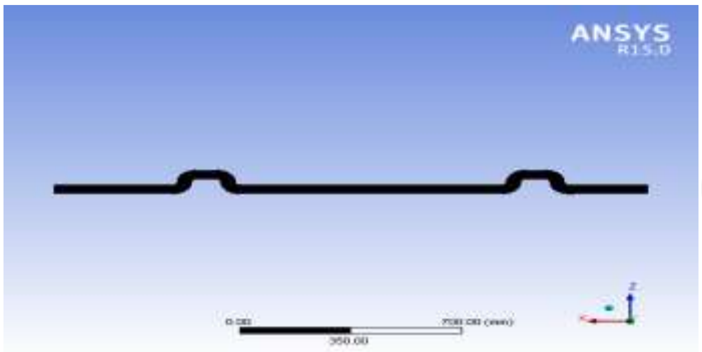


Figure 6 Meshing of crank shaft 40 mm diameter

10. Static structural Analysis of the existing component with diameter 40 mm and crank web 44mm

A) Von-Mises stress
The maximum stress occurred in the component is 71.293MPa

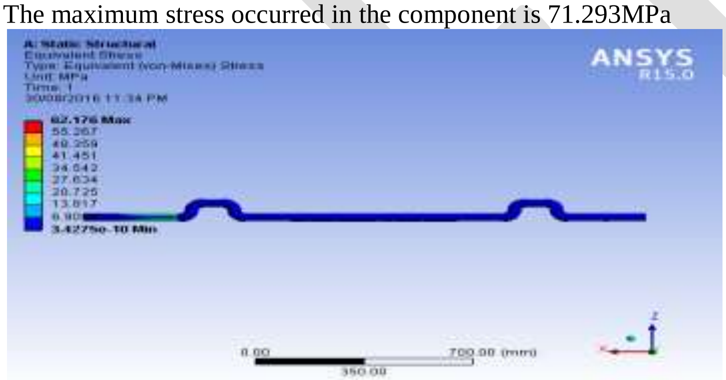


Figure 7 Von Mises stress of crank shaft 40 mm diameter

B) Deflection
Deflection occurred in the component is 0.32239mm

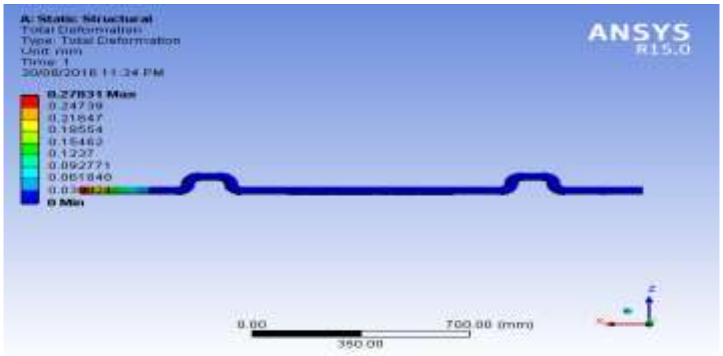


Figure 8 Deflection of crank shaft 40 mm diameter

Table 3

Model	Material	Weight in kgs	Max Deflection in mm	Von-mises Stress in Mpa
Existing Crank shaft 40 mm diameter with crank web 44mm	Cold drawn steel	20.658	0.3223	71.29
Crank shaft 40 mm diameter with crank web 46mm	Cold drawn steel	20.953	0.3141	73.14
Crank shaft 38 mm diameter with crank web 42mm	Cold drawn steel	18.334	0.3946	90.29
Crank shaft 38 mm diameter with crank web 44mm	Cold drawn steel	18.820	0.3843	76.922
Crank shaft 42 mm diameter with crank web 48mm	Cold drawn steel	23.115	0.4335	63.20
Crank shaft 42 mm diameter with crank web 46mm	Cold drawn steel	21.992	0.4335	77.11
Crank shaft 44 mm diameter with crank web 48mm	Cold drawn steel	25.452	0.4647	49.027

Static structural analysis for different diameters with different wed radii are carried, the results are shown in the table 3. From the above results table it can be observed that the diameter of crank shaft is 40 mm with web diameter is 44 mm and weight of the component is around 20.658 kg. After static structural analysis it was observed that the maximum stress and deflection values are 71.29 MPa and 0.3223 mm respectively.

After redesigning and modeling the component for different diameters and weights, it is observed that the crank shaft of diameter 44 mm with 48 mm crank web having a weight of 25.452kg, the stress and deflection values are 49.027 MPa and 0.4647 mm. The stress obtained from modified component is less than the existing component. Hence it can be considered as safer design.

Table 4

Sl No	Mode	Modal Frequency for Φ 40mm crank shaft with web 44	Modal Frequency for Φ38mm crank shaft with web 42mm	Modal Frequency for Φ42mm crank shaft with web 48mm	Modal Frequency For Φ44mm crank shaft with web 48mm
1	1	67.818	48.046	51.674	53.882
2	2	68.848	48.909	52.741	54.767
3	3	72.177	71.795	74.258	78.918
4	4	73.225	72.9	75.626	80.156
5	5	333.19	258.71	285.15	292.63
6	6	339.9	259.65	285.29	293.22

From the above table 4 it is observed that the minimum frequency obtained from the existing component is around 67.818Hz, which is greater than the operating frequency of 2Hz hence the component is more safer.

After redesigning and modelling the component for different diameter and weights it is observed that the crank shaft of diameter 44 mm having a weight 25.452 kg, the minimum frequency obtained is 53.882 Hz, which is also greater than the operating frequency, which is found to be safer design.

The dynamic results obtained from the numerical analysis are validated with the experimental results by using FFT analyzer.

11. Conclusions

In this work the component is designed by considering the loads acting on it. Static and dynamic analysis of the component is carried out by using FEA. In the existing crank shaft high stresses were found at the crank web region due to sudden change in cross section area, which was the main cause for failure of the crank shaft for failure.

Then the existing crank shaft is modelled for different diameters with different web radii then numerical analysis is carried out using FEA software. It is observed that the crank shaft of diameter 44 mm has the stress 49.027 MPa which is lesser than the stress occurred in the existing component and modal frequency 53.882Hz value which is greater than the operating frequency for a weight of 25.452 kg. Hence this component is considered as safe.

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