Investigation of Wear Characteristics of Non-Edible and Mineral Oils with CNT additive in Precision Transmission Chain Pins

C.G. Pradip Udayakanth[1], D.Pavithira[2] and Dr.P.Sadagopan[3,*]
Dept. of Production Engineering, PSG College of Technology, Coimbatore

Abstract

From time immemorial, after the invention of wheels many technological developments have taken place in machinery, equipment and in automobile applications. The mechanisms have been continuously developed for transmitting motion, power. However, the mechanisms developed are found to have limited life due to failure caused by fatigue and wear and sometimes due to overloading. Many research works have been carried out to enhance the life of mechanisms by reducing the wear and improving the fatigue life. Many researchers reported that the band width of life or the consistency and improvement in life is possible by close control of dimensions by precision manufacturing. In addition, oils are found to be very useful in operating machines smoothly and understanding of their physical and chemical properties led to their novel uses in the industrial world. The wear of cylindrical pins similar to chain pins used in precision transmission depends on surface topography influenced by surface finish, surface geometry, relative hardness of mating parts, surface coating etc and the lubricants and the method of application of lubricants. In any transmission system, all these parameters influence wear, thereby causing increased play or change in centre distance or length. In the case of transmission chain, the above said parameters cause elongation due to the cumulative effect of wear of pins and bushes. The present work concentrates on the lubricating capacity of different types of non-edible seeds like Annona squamosa, Vitis vinifera and Heveabrasiliensis, etc and testing their wear characteristics in comparison with mineral oil for chain pins. It is found that the bio-oils except Annona squamosa have better wear characteristics in comparison with standard SAE 40, 90 mineral oils. In addition, the effect of the addition of carbon nano tubes to the SAE 40 oil, in chain pin wear is studied and the results are discussed.

Keywords: Chain pins, Bush, Wear, Elongation, Lubricants, Carbon nano tubes(CNT), Mineral oil, Bio-oil, Pin on disc.

1. INTRODUCTION

Chains are used extensively in the final drive transmission and in timing application of four stroke single cylinder engine of motorcycle. In additions chains are used in many industrial applications viz. as a drive, conveyor, elevator, escalators, etc. Even though chains have only four components which get repeated in numerical terms depending on its length, it is manufactured with very close tolerance and precisely assembled. In any transmission or mechanism, wear is inevitable due to the relative motion of its components. In a similar fashion in a transmission chain, wear and hence chain elongation is inevitable. Wear of chains in precise timing application would cause a change in the timing and hence cause engine misfiring. Motorcycle drive chain elongation is mostly due to wear of chain pins, bushes and deformation of its components due to the excessive load caused by abused driving or overloading the vehicle which is common in our country. Also, fatigue loading causes chain elongation. A roller chain is known to be one of the positive transmission systems owing to the absence of slip. Compared with that of belts, the load/power transmission capacity of a roller chain is large and the efficiency is high. However, because of polygonal action during meshing, vibration and noise are found to be high between 72dB to 84 dB as reported by Zheng, et al [1]. In order to reduce vibration and noise in chain transmission, silent chains have been developed. Silent chains are mostly used in engine timing and front wheel drive applications in cars. Radcliffe [2] studied wear mechanism in unlubricated articulated chains. Peeken and Coenen [3] analysed the influence of different lubrication conditions on the wear of roller chains.

Fawcett and Nicol [4] used a discrete dynamic method to develop a dynamic chain model and reported that load on sprocket tooth varied with a change in pressure angle and number of teeth. Naji and Marshek [5] analysed the effect of pressure angle in a chain drive and reported the variation due to pitch difference between chain link and sprocket teeth. Sadagopan, et al[6] theoretically analysed the elongation and fatigue of drive chains used in 100 cc motorcycles and reported that Archard's model could be used for chain elongation estimation and by modifying bush inner profile, increased life of chain could be achieved. Kerremans [7] studied wear on conveyor chains with polymer rollers. James *et al* [8] analysed the effects of frictional loss on bicycle chain drive efficiency analytically and experimentally. Wang *et al.*[9] made a study on jumping over teeth phenomenon in roller chain drive. Sprocket and chain noise investigation was carried out by Liu *et al.* [10] and the authors reported that meshing sound pressure is associated with chain speed. Contact pressure and load distribution between steel roller chain and polymer sprocket were reported better due to deflection of sprocket teeth[11]. A model of chain and sprocket with chain guide for marine diesel engine was made and reported polygonal effect and link force variations [12-13]. Pitch variations and friction between sprocket and chain roller cause higher load. Due to wear, roller teeth of sprockets are subjected to a higher load level than pin teeth [14].Conwell and Johnson [15] investigated experimentally tension in links and impact of rollers on a sprocket. Sadagopan*et al.* [16] presented the effect of assembly variations in the fatigue life of motorcycle chains both by simulation and by the experimental method. Nikhil *et al.* [17] made FEA analysis on motorcycle chains using ANSYS software. Pereira *et al.* [18] made a study of chain mechanism using clearance revolute joint between roller and sprocket with contact condition as a function of sprocket tooth profile.

*Author email for correspondence: sadagopan.p@gmail.com

ISBN: 978-93-80689-28-9

In motorcycles, roller chains are used in the final drive that connects the drive sprocket mounted on the output shaft of gear box and the driven sprocket fitted to rear wheel. The rear wheel axle is adjustable to reduce slackness in chains owing to elongation caused mostly by wear of pins and its deformation and that of wear of bushings. The life span of the roller chain depends on its elongation and normally limited to 2 to 3% of the initial length. Improper alignment of drive and driven sprockets while fitting in motorcycle may lead to increased elongation, higher noise and reduced efficiency of chain drive.

The transmission chains used in motor cycles and mopeds are mostly with 12.7mm and 15.875mm pitches. These chains are mass-produced and there are assembly variations noticed due to improper assembly of components, dimensional tolerances, heat treatment distortion, improper burr removal in plates etc. Most investigations on chain performance are done in classical four square test benches [19]. The main problem associated with four square test is that only cumulative wear of chains can be determined and not the wear of individual pins. Moreover, in four square tests, chain sprocket alignment variations, manufacturing variations like pitch, geometry etc., in both chains and sprockets may lead to non uniform load distribution in chain links causing variations in wear of pins and bushings.

The factors that influence chain pin wear are relative hardness of pins and bushes, surface finish, shrinkage of bush owing to interference fit, skew assembly of pins and bushes with respect to outer and inner plates, preloaded level of chains and the driving conditions. Above all, the type of lubricants and the method of application of lubricants between pin and bush have greater influence on chain pin and bush wear [20]. In this paper an attempt is made to find the chain pin wear in dry run as well as using different lubricants. It is found that the lubrication has the most influential factor for chain pin wear that causes chain elongation.

2. ROLLER CHAIN

A roller chain is made up of a number of outer and inner links fitted with pins, bushes and rollers as shown in the Fig.1. The dimensions as per DIN 8187 of 15.875 mm pitch are shown in Table 1.Pins and bushes are made of low carbon, low alloy steel and are case hardened and polished. Plates, both inner and outer are made of medium carbon steel and through hardened and shot peened to improve fatigue strength. These components are made in special purpose machines and in huge volumes as per ISO/DIN standards. The chains are made in automatic assembly machines and preloaded and lengths are measured under standard loads as per roller chain standards. In order to improve the wear characteristics, chain pins can be coated to a higher hardness by chromizing diffusion heat treatment or by hard chrome plating. These processes increase the cost of chains to a much higher level and it will not be economical to sustain in the highly competitive market. Hence, improvement in assembly and close control in dimensional or geometric characteristics and improved surface finish can lead to a better life of chains in addition to the lubricants used.

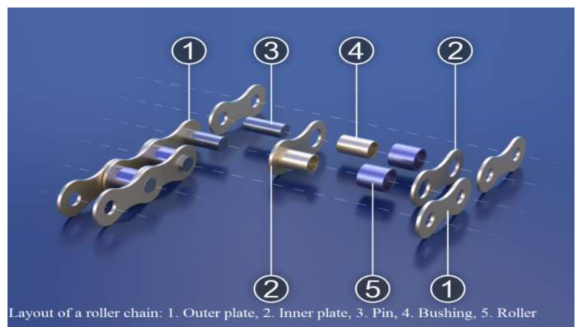

Fig.1. A Roller Chain

Table 1. Chain Link Dimensions

ISO No.	Pitch P (mm)	Maximum Roller Diameter R (mm)	Maximum Pin Diameter P (mm)	Maximum Pin Length P_l (mm)	Minimum Width Between Inner Plates WBIP (mm)	Inner & Outer plate thickness (mm)
10B-1	15.875	10.16	5.08	18.8	9.65	1.55

Fig.2.Schematic diagram of normal position of pin on disc

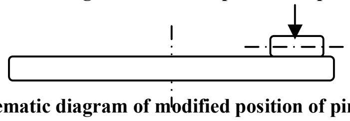

Fig.3.Schematic diagram of modified position of pin on disc

3. CHAIN PIN WEAR ANALYSIS

In this work, the wear behaviour of chain pin was tested using the pin on disc tribometer. Normally in a pin on disc experiments, pin ends are tested for wear with a pin end pressed against a rotating hardened and tempered steel disc with a hardness level >58 HRC as shown in Fig. 2.Chain pins are case hardened and only the pin body gets worn out due to articulation movement inside the bush and moreover, the pin ends are riveted. Hence, the normal method of the pin on disc wear study will not reflect the wear behaviour of chain pins. Hence, in this work, in order to study the wear of pin body, a niche fixture is designed and developed and the pins are mounted in the fixture so that the pin body which is in fixed condition gets in contact with the rotating disc as shown in Fig.3.The fixture is fitted to the loading arm as shown in Fig.4.

3.1 Theoretical estimation of pin wear

The theoretical evaluation of chain pin wear is carried out using Archard's wear model as shown in equation (1). Initially, chain pin wear is estimated based on the assumption of Archard's wear constant as $5X10^{-3}$[6] in the dry run. In this model since the pin is fixed, the sliding distance is considered equal to the experimental sliding distance due to disk rotation and the load is equal to the applied load in pin on disc tribometer. Yield strength value is taken as proportional to the pin hardness level. The projected area of pin is found by multiplying pin diameter and width of pin used for testing.

$$\Delta = (k*F*L*1000)/(9S_y*A) \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

where Δ is wear in mm/m, k is Archard's wear coefficient, F is load in N, S_y is yield strength N/mm^2, L is length of sliding in m and A is projected bearing area of pin in mm^2.

The theoretical wear of pins is calculated with Archard's wear coefficient $5X10^{-3}$ for unlubricated condition.

A=5.08*18;F=50N;L=1782m;2591m;5184m.

Based on the theoretical wear, the wear volume , its weight and wear % are estimated using equation (2) and tabulated as shown in Table 2.

$$\left.\begin{array}{l} \textit{Wear volume } v=\{((\pi r^2\theta/_{180})-r\sin\theta(r-\Delta))l\}\ mm^3 \\ \textit{Wear mass}= v\rho\ gm \\ \textit{Wear in \%}= (\textit{wear mass/theoretical mass})\ X100 \end{array}\right| \ldots\ldots (2)$$

3.2 Experimental study

Chain pins were removed from the finished chain and inspected for hardness, surface finish etc and are tabulated as shown in Table 3. The sliding time to be set in control panel was calculated from the known value of sliding distance and sliding velocity. A constant load of 50N was applied by placing weights on the loading arm. The wear of the chain pin was studied as a function of the applied load, sliding velocity, sliding distance, with and without lubrication and are shown in Table 5.

Fig.4 Pin on disc with a fixture for positioning chain pin

Table 2 Theoretical mass loss of chain pins in unlubricated condition

Pin	1	2	3	4
Hardness HV_1	798-819	781-795	779-795	779-795
Strength MPa	2040	2040	2040	1400
Load N	50	50	50	50
Sliding distance	1782	2590	5184	5184
Wear height	0.265	0.386	0.772	1.124
Wear volume mm^3	0.394	0.704	1.943	3.324
Wear mass gm	0.003	0.0055	0.015	0.026
Wear in %	0.104	0.192	0.523	0.907

Table 3 Hardness, finish and dimension of chain pins

S.no	Hardness HV	Surface roughness Ra	Outer diameter Mm
1	779-795	0.37	5.077-5.079
2	725-745	0.52	5.079-5.084
3	798-819	0.46	5.078-5.080
4	726-786	0.46	5.072-5.076
5	781-795	0.59	5.070-5.074

Similarly, the wear in percentage is found by conducting pin on disk method using mineral oils SAE 40, SAE 80, bio-oils Annona squamosa, Vitis vinifera, Heveabrasiliensisand the results are tabulated as shown in Table 6. The properties of mineral oils and bio-oils are shown in Table 4.

Table 4 The physical properties of mineral oils and bio-oils

S.no	Property	Annona squamosa	Vitis vinifera	Heveabrasiliensis
1	Specific gravity (g/cm^3)	1.01	0.83	0.90
2	Viscosity (cSt)	38.6	121	109
3	Pour point °C	4	4	6
4	Flash point °C	230	320	210
5	Fire point °C	260	380	240

4. RESULTS AND DISCUSSION

From the theoretical estimation of wear which is shown in Table 2, it is observed that the wear increases with increase in sliding distance. The theoretical wear mass values which are shown in columns 1 and 2 of Table 2 closely agree with the experimental results shown in Table 5.However, theoretical wear mass shown in column 3 of Table 2 does not agree with the experimental results shown in Table 5. This may be due to the assumption of constant yield strength. As the pin wears out, the surface hardness decreases until the core is reached. The hardness of the core is much lower which is of the order of 400 HV to 500HV with a mean value of 450 HV that correspond to 1450N/mm^2[21].Using this value, the theoretical wear is calculated as shown in column 4 of Table 2. It is observed that the revised estimate of wear mass closely agrees with the experimental results in unlubricated condition shown in Table 5.

From the values of Tables 2 and 5, a plot is made as shown in Fig.5. From the plot it is observed that the wear of pins under unlubricated condition steadily increases as the sliding distance increases. Theoretical wear estimate with constant yield strength of 2040N/mm^2 closely agrees with the experimental values up to a sliding distance of 2590m beyond which it is lower. By modifying the yield strength beyond sliding distance of 2590m, to lower level of 1450N/mm^2 based on lower hardness near the core, it is found that the theoretical estimate closely agrees with the results obtained by the experiment. Also, it is observed that the pin lubricated with CNT 10gm in 100ml SAE 40 oil has very low wear in comparison with the unlubricated condition.

Table 5 Experimental wear test results with no lubrication and CNT lubrication

S.no	Load N	Sliding velocity m/s	Sliding distance m	Mass before test (gm)	Mass after test (gm)	Mass loss (gm)	Wear in percent	Average Frictional force N	Lubrication
1	50	2.88	1728	2.9205	2.9164	0.0041	0.14	28	No lube
2	50	2.88	2591	2.93	2.9245	0.0055	0.188	26	-do-
3	50	2.88	5184	2.9028	2.8787	0.0241	0.83	28	-do-
4	50	2.88	2590	2.9251	2.9246	0.0005	0.017	3	CNT
5	50	2.88	5184	2.9293	2.9276	0.0017	0.058	2	CNT
6	50	2.88	7775	2.9243	2.9204	0.0039	0.133	2.5	CNT

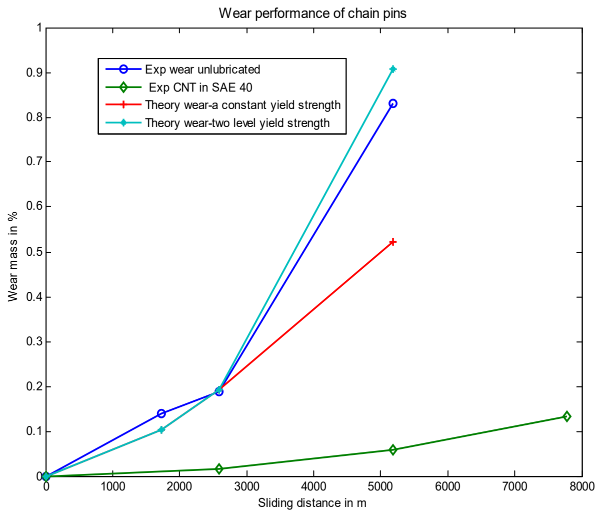

Fig.5 Theoretical and experimental wear test plots for chain pins

Typical worn pin obtained by Pin on disc wear test with a fixture attached to loading arm is shown in Fig.6

Fig.6 Wear pattern on the body of chain pin

From the Table 6 and also from the plot as shown in Fig.7, it is observed that the pins lubricated with Vitisvinifera bio-oil have the lowest wear in comparison with Annona squamosa and Heveabrasiliensis and both mineral oils SAE 40 and SAE 80 oils. It is also observed that the bio-oils Vitisvinifera andHeveabrasiliensis have better lubrication properties in comparison with mineral oils.

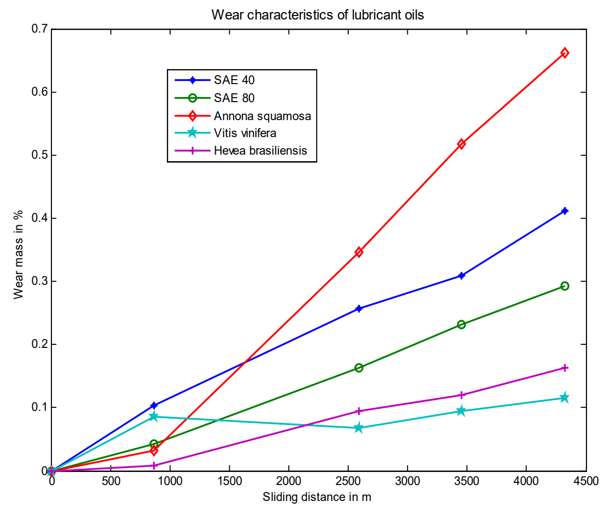

Fig.7 Comparison of wear performance of chain pins in mineral and bio-oils

6. CONCLUSIONS

1. In order to study the wear of chain pins on the surface of the body due to articulation, a niche fixture is designed and fabricated to keep the pin flat as against conventional method of pin orientation with respect to the disk.

2. Theoretical wear of chain pins were estimated using Archard's wear model in unlubricated condition and compared with the experimental values and found that theoretical estimate closely agrees with experimental results.

Table 6 Pin on disc wear test results for chain pins using mineral and bio-oils

		Wear of pins in percentage				
Time	Sliding distance	SAE 40	SAE 80	Annona squamosa	Vitis vinifera	Heveabrasiliensis
5	864	0.103	0.043	0.032	0.086	0.008
15	2591	0.257	0.163	0.347	0.068	0.094
20	3455	0.309	0.232	0.518	0.094	0.123
25	4320	0.412	0.292	0.662	0.115	0.163

3. Mineral oils and bio-oils wear performance was studied and observed that bio-oils Vitis vinifera, Heveabrasiliensis performed better in comparison with minerals oils. Moreover, Vitis vinifera bio-oil has better wear characteristics than the other bio-oils.

4. Wear performance of CNT oils is compared with the pins in unlubricated condition and also compared with other oils against sliding distance and found that the CNT has good performance with 10gm of CNT in 100ml SAE 40 oils. However, the effect of varying the percentage of CNT has to be carried out in future. Also, the wear characteristics of other bio-oils have to be carried out.

5. The effect of surface finish, geometric characteristics of pins have to be carried out in future.

References

[1] Zheng H,WangY.Y,Liu G.R and Lam K.Y, Quek K.P , Ito T and Noguchi Y, Efficient modeling and prediction of meshing noise from chain drives, *Journal of sound and Vibration*,245(1) (2001)133-150.

[2] Radcliffe S.J., Wear mechanisms in unlubricated chains, *Tribology International*, 14(5) (1981) 263-269.

[3] Peeken,H.Coenen,W Influence of oil viscosity and various additives on the wear of roller chains, *Wear,* 108 (4) (1986) 303-321.

[4] Fawcett J N & Nicol S W, A Theoretical investigation of the vibration of roller chain drive, *Proc Fifth World congress on Theory of Machines & Mechanisms (ASME, Montreal)* (1979)1482-1485.

[5]Naji M R and Marshek K M, Experimental determination of the roller chain load distribution, *ASME J Mech. Trans Auto Des*, 105 (1983) 331-338.

[6] Sadagopan P, Rudramorthy R and Krishnamurthy R, Wear and fatigue analysis of two wheeler transmission chain, *Journal of Scientific & Industrial Research*, 66 (2007) 912-918.

[7] Kerremans,VRolly,. De Baets T.P, DePauw J, Sukumaran J and Perez Delgado Y, Wear of conveyor chains with polymer rollers, *Sustainable Construction and Design*,(2011) 378-387.

[8] James B S, Christopher J K R, Michael J E, Johanna R B, Masahiko F & Masao T, Effects of frictional loss on bicycle chain drive efficiency, *Transaction ASME,* 123 (2001) 598-605.

[9] Wang Y, Zheng Z, Zhang G, A Study on jumping-over-teeth phenomenon in roller chain drive, *ASME J Mechanical Design*, 112 (1990) 569-574.

[10] Liu S.P, Wang K.W, Hayek S I, Trethewey M W and Chen F H K,A global local integrated study of roller chain meshing dynamics, *Journal of Sound and Vibration*, 203(1997) 41-62.

[11] Eldiwany B.H and Marshek K.M, Experimental load distribution for double pitch steel roller chain on polymer sprockets, *Mechanism and Machine Theory*,24 (5) (1989) 335-349.

[12] Pedersen S.L, Hansen J.M, and Ambrosio J.A.C, A roller chain drive model including contact with guide bar , *Multibody System Dynamics,*12 (2004) 285-301.

[13] Pedersen S.L, Model of contact between rollers and sprockets in chain drive system, *Archive of Applied Mechanics*, 74 (2005) 489-508.

[14] Naji M.R and Marshek K.M, The effects of pitch difference on the load distribution of a roller chain drive, *Mechanism and Machine Theory*,24 (1989) 351-362.

[15] Conwell J.C and Johnson G.E, Experimental investigation of link tension and roller sprocket impact in roller chain drives, *Mechanism and Machine Theory*, 31 (1) (1995) 533-544.

[16] Sadagopan P, Harish V, Pradip Udayakanth C G, Fatigue analysis of motorcycle chains, *International Journal of Applied Engineering Research*, 23(2014)19145-19176.

[17] Nikhil S. Pisal, V.J. Khot, Swapnil S. Kulkarni, Design and development of motorcycle chain links by using C.A.E software, *International Journal Scientific Research and Management studies*,2(4) (2015) 175-183.

[18] Pereira C, Ambrósio J and Ramalho A, Dynamics of chain drives using a generalized revolute clearance joint formulation, *Mechanism and Machine Theory*, 92 (2015) 64-85.

[19] Conwell J.C and Johnson G.E, Design and construction and instrumentation of a machine to measure tension and impact forces in roller chain drives, *Mechanism and Machine Theory*, 31 (4) (1996) 525-531.

[20]Lee P.M and Priest M, An innovation integrated approach to testing motorcycle drive chain lubricants, *Proceedings of 30th Leeds-Lyon symposium on Tribology*, *Tribology and interface Engineering series, Elsevier,* (2004) 291-298.

[21] www.globalmetals.com.au