

**SUDAN UNIVERSITY OF SCIENCE AND TECHNOLOGY**

**FACULTY OF GRADUATE STUDIES**

**CYLINDER'S LINER WEAR  
CHARACTERISTICS IN TWIN CRANKSHAFT  
INTERNAL COMBUSTION ENGINES**

**خواص التآكل في مبطنات أسطوانات محركات الاحتراق الداخلي ثنائية  
عمود المرفق**

By

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# Abstract

Conventional internal combustion engines have been around for more than a century. Due to its high efficiency and low cost.

The overall efficiency of IC engines approaches 50%. This means that only about 50% of the fuel is converted into useful work, and the rest of fuel power is useless energy mainly due to friction between moving parts.

Friction is the worst enemy of machinery. It wears out metal, wastes power and generates heat. Friction occurs in all moving parts in the engine, like piston rings, journal bearings, valves and cams. Friction between piston rings, cylinders liner responsible for about half of the usefulness energy <sup>(1)</sup>.

Sliding contacts between a piston ring and a cylinder liner leads to a variety of different friction mechanisms during one working cycle of the engine. Owing to the variations in load, speed and counter surface effects, lubrication conditions in a ring/liner contact are strongly transient, which is reflected by variations in the friction and wear behavior.

Wear in cylinder liner causes inefficiency seal, and it may result in increased oil consumption, leakage, blow-by, and increasing fuel consumption.

Combustion pressure cause a force on piston, the horizontal component is a side thrust force which is normal to liner wall. This force is responsible for the friction between piston ring and wall and then wear of cylinder liner. Beside the minor causes by piston ring surfaces finishing and coating, cylinder liner material, cylinder liner surface finishing and coating and cylinder liner out of roundness.

The aim of the present work is to minimize side thrust force as it is major cause of cylinder liner wear.

In the present work an arrangement of twin crankshaft model has been manufactured in Sudan University of Science and Technology-College of Engineering as a solution attempt to overcome cylinder friction and then reducing cylinder wear.

Twin crankshaft engine uses two contra-rotating crankshafts, geared together, and connected to a piston through two connecting rods driven by one crankshaft.

Experimental work was carried out to investigate wear characteristics using three different engines arrangement inline, offset and twin crankshaft engines, for two engine's speeds.

A computer program was used to model the three engines arrangements and to obtain their theoretical performances.

A comparison between the experimental results of the three engines arrangement showed that twin crankshaft engine arrangement is a promising solution for reduction of cylinder liner wear. This is in agreement with the predicted results obtained by the computer program.

## الخلاصة

محركات الاحتراق الداخلي التقليدية لا تزال تستخدم لأكثر من قرن. بسبب كفاءتها العالية وكلفتها المنخفضة.

الكفاءة الكلية لهذه المحركات حوالي 50%. هذا يعني بأن حوالي 50% فقط من طاقة الوقود يتحول إلى عمل مفيد، والباقي عبارة عن طاقة مهدرة بسبب الاحتكاك بين الأجزاء المتحركة.

الاحتكاك أسوأ عدو للماكينات إذ أنه يُضعف المعدن ويهدر القدرة ويولد الحرارة. يحدث الاحتكاك في كل الأجزاء المتحركة في المحرك، مثل حلقات المكبس والمحامل والصمامات والحدببت، لكن الاحتكاك بين حلقات المكبس ومبطن الأسطوانات مسئول عن حوالي نصف الطاقة الضائعة في الاحتكاك.

التلامس المنزلق بين حلقات المكبس ومبطن الأسطوانة يُؤديان إلى حدوث آليات احتكاك مختلفة أثناء دورة عمل واحدة من المحرك. وبسبب الاختلافات في الحمل، السرعة، التأثيرات السطحية، ظروف التزييت بين حلقات المكبس ومبطن الأسطوانة تكون عابرة، الأمر الذي يؤدي إلى تفاوت في حدوث التآكل والاحتكاك.

التآكل في مبطن الأسطوانة يؤدي إلى تقليل كفاءة حلقات المكبس التي تعمل كمانعة للتسرب، وهذا قد يؤدي إلى استهلاك متزايد للزيت، تسرب غازات غرفة الاحتراق وبالتالي تقليل ضغط الاحتراق، واستهلاك متزايد للوقود.

يسبب ضغط الاحتراق قوة على المكبس، المكون الأفقي لهذه القوة هو قوة الدفع الجانبية التي تؤثر عمودياً على سطح المبطن. هذه القوة مسئولة عن الاحتكاك بين حلقات المكبس ومبطن الأسطوانة ومن ثم تؤدي إلى تآكل مبطن الأسطوانة. إلى جانب الأسباب البسيطة الأخرى مثل صقل وطلاء سطوح حلقات المكبس، مادة مبطن الأسطوانة، صقل و طلاء مبطن الأسطوانة و الأسطوانة غير مكتملة الاستدارة.

إنّ هدفَ العملِ الحاليّ هو تخفيضُ قوّةِ الدّفعِ الجانبيّةِ التي هي سببُ رئيسيُّ لتآكلِ مبطنِ الأسطوانة.

في العملِ الحاليّ تمّ تصنيعُ نموذجٍ لماكيّنة ثنائيّة عمود المرفق في جامعةِ السودانِ للعلومِ والتكنولوجيا - كلية الهندسة لمحاولةِ لحلِّ مشكلةِ الاحتكاكِ بينِ المكبسِ و الأسطوانة وبالتالي تخفيضِ تآكلِ الأسطوانة.

المأكيّنة ثنائيّة عمود المرفق هي عبارة عن نظامٍ يستخدمُ عمودي مرفق يدوران في اتجاهين متعاكسين ويتصلان بالمكبس عن طريق ذراعي توصيل يتم تدويرهما بواسطة عمودي المرفق.

تمت كتابة برنامج حاسوبي لنمذجة الترتيبات الثلاثة للمحركات وللحصول على الأداء النظري لها.

العمل التجريبي نُفذَ لثلاثة ترتيباتٍ لمحركاتٍ مختلفة: المأكيّنة التقليدية والمأكيّنة ذات عمود المرفق اللامتمركز وأخيراً المأكيّنة ثنائيّة عمود المرفق. عند سرعّتين مختلفتين للمحرك.

أجريت مقارنة بين النتائج المتحصلة للمحركات الثلاثة ووجد أن المأكيّنة ثنائيّة عمود المرفق تعدُّ حلاً واعدًا لتخفيضِ تآكلِ مبطنِ الأسطوانة. هذه النتائج جاءت متوافقة مع نتائج البرنامج الحاسوبي.

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# List of symbols:

<b>a</b>	Piston acceleration, $m/s^2$
<b>A<sub>bc</sub></b>	Rod's small end loading area, $m^2$
<b>A<sub>N-N</sub></b>	Shank cross-section area, $m^2$
<b>A<sub>p</sub></b>	Cylinder cross-section area, $m^2$
<b>b</b>	Connecting rod length, m
<b>c</b>	Cylinder diameter, m
<b>C</b>	Static load, $N/m^2$
<b>C<sub>b</sub></b>	Belt center distance, m
<b>C<sub>F</sub></b>	Surface condition factor
<b>C<sub>p</sub></b>	The elastic coefficient, $\sqrt{\text{psi}}$
<b>C<sub>v</sub></b>	Specific heat, (kJ/kg.k)
<b>D</b>	Cylinder diameter, m
<b>d</b>	Crankshaft offset from the axis of piston movement, m
<b>D<sub>bp</sub></b>	Outer diameter of the main pin, m
<b>D<sub>i</sub></b>	Inner diameter of the main pin, m
<b>D<sub>ip</sub></b>	Inner diameter of big end pin, m
<b>D<sub>is</sub></b>	Rod's big end inner diameter, m
<b>d<sub>op</sub></b>	Rod's big end outer diameter, m
<b>d<sub>os</sub></b>	Rod's small end outer diameter, m
<b>d<sub>p</sub></b>	Pitch diameter, m
<b>D<sub>PL</sub></b>	Outer diameter of the big end pin, m
<b>F<sub>n</sub></b>	Net face width, m
<b>F</b>	Force applying on the end of the crank arm, N
<b>F<sub>(thrust)</sub></b>	Side thrust force, N
<b>F<sub>f</sub></b>	Friction force, N
<b>F<sub>g</sub></b>	Gas force, $N/m^2$
<b>f<sub>i</sub></b>	Inertia force, N
<b>fmep</b>	Friction mean effective pressure or work to overcome friction, atm
<b>F<sub>r</sub></b>	Rod applied force, N
<b>F<sub>s</sub></b>	Piston force, N
<b>f<sub>s</sub></b>	Safety factor
<b>g</b>	Gravity, $m/s^2$
<b>h</b>	Horsepower

<b>H</b>	Life time for the bearing, hours
<b>K<sub>B</sub></b>	Rim-thickness factor
<b>K<sub>m</sub></b>	Load distribution factor
<b>K<sub>o</sub></b>	Overload factor
<b>K<sub>s</sub></b>	Shape factor
<b>K<sub>v</sub></b>	Dynamic factor
<b>L</b>	The length between the centers of the big and small pins, m
<b>L<sub>p</sub></b>	Rod's big end length, m
<b>L<sub>PL</sub></b>	The length of the big end pin, m
<b>L<sub>s</sub></b>	Rod's small end length, m
<b>m</b>	Mass of the piston and piston's pin, kg
<b>mmep</b>	Work required to motor an engine, atm
<b>n</b>	Constant
<b>N</b>	Force applying on the piston, N
<b>P</b>	Gas pressure, N/m <sup>2</sup>
<b>P<sub>h</sub></b>	Power produced by the electric motor, W
<b>P<sub>b</sub></b>	Dynamic load applying on bearings, N
<b>P<sub>d</sub></b>	Diametric pitch, (1/in)
<b>Q</b>	Heat addition, kJ
<b>r</b>	Radius of crankshaft, m
<b>R</b>	Gas constant, (kJ/kg.k)
<b>R<sub>p</sub></b>	Radius of the big pulley, m
<b>r<sub>p</sub></b>	Radius of the small pulley, m
<b>T</b>	Temperature, C°
<b>t<sub>s</sub></b>	Rod's small end thickness, m
<b>v</b>	Velocity, m/s
<b>V</b>	Volume, m <sup>3</sup>
<b>x</b>	Fraction of heat release

## **Greek symbols:**

<b><math>\alpha</math></b>	Angle between the extension of connecting rod and the line perpendicular to the crank arm, degrees
<b><math>\gamma</math></b>	Specific heat ratio
<b><math>\theta</math></b>	Angle between the crank and the line joining the center of the crankshaft to the piston, degrees



$\mu$	Friction factor
$\sigma_b$	Load stress, $\text{N/m}^2$
$\sigma_c$	Contact stress for the gear, $\text{N/m}^2$
$\sigma_{cc}$	Compression stress, $\text{N/m}^2$
$\tau_{\max}$	Maximum shear stress, $\text{N/m}^2$
$\phi$	The angle between the connecting rod and the axis of piston movement, deg
$\omega$	Angular velocity, rad/s

## Subscripts:

<b>b</b>	bearing
<b>b</b>	belt
<b>c</b>	contact
<b>d</b>	diameter
<b>g</b>	gas
<b>i</b>	inner
<b>ip</b>	Inner, big
<b>max</b>	maximum
<b>n</b>	net
<b>o</b>	overload
<b>op</b>	Outer, big
<b>p</b>	pulley
<b>r</b>	rod
<b>s</b>	small

# Table of content:

<b>Abstract</b>	II
<b>Arabic abstract</b>	IV
<b>Acknowledgment</b>	VI
<b>List of symbols</b>	VII
<b>Chapter one</b>	<b>1</b>
<b>Introduction</b>	1
1.1 Preliminary remarks	2
1.2 Statement of problem	3
1.3 Scope and subject of present work	6
1.4 Objectives of present work	7
<b>Chapter two</b>	<b>9</b>
<b>Liners wear, Causes, effects and cures</b>	9
2.1 Introduction	10
2.2 Factors affecting wear in engine's cylinder liners	12
2.2.1 Friction	12
2.2.2 Engine speed	14
2.2.3 Piston ring surface finishing and coating	15
2.2.4 Liner material	16
2.2.5 Cylinder liner surface finishing and coating	16
2.2.6 Side thrust force	17
2.3 Computational simulation	19
2.4 Effect of wear in engine's cylinder liners on engines performance	21
2.4.1 Combustion gases (blow-by)	21

2.4.2	Oil consumption	22
2.4.3	Noise and vibrations	22
2.5	Remedies of wear	22
2.5.1	Lubrication	23
2.5.2	Piston specifications	24
2.5.3	Piston rings	25
2.5.4	Rotating liner engine	28
2.5.5	Offset wristpin	28
2.5.6	Offset crankshaft	29
2.5.7	Revetec engine	30
2.5.8	Twin crankshaft engine	30
2.5.8. 1	Connecting rods affixed to one wrist pin	33
2.5.8..2	Connecting rod affixed to two spaced wrist pins	33
2.5.8. 3	Arced connecting rods	33
2.5.8. 4	Connecting rods in crossed configuration	34
	<b>Chapter three</b>	<b>37</b>
	<b>Theoretical approach</b>	<b>37</b>
3.1	Introduction	38
3.2	Computer program	39
3.2.1	Program flow chart	40
3.2.2	Piston displacement, speed and acceleration	42
3.2.2.1	Inline crankshaft engine	42
3.2.2.2	Offset crankshaft engine	43
3.2.2.3	Twin crankshaft engine	45
3.2.3	Engines performance	45
3.2.3.1	Inline crankshaft engine performance	46

3.2.3.2	Offset crankshaft engine performance	51
3.2.3.3	Twin crankshaft engine performance	53
3.2.4	Theoretical torque	54
3.2.4.1	Inline crankshaft engine performance	57
3.2.4.2	Offset crankshaft engine performance	59
3.2.4.3	Twin crankshaft engine performance	60
<b>Chapter four</b>		<b>63</b>
<b>Experimental test rig</b>		<b>63</b>
4.1	Introduction	64
4.2	Design of test rig	64
4.2.1	Connecting rod	65
4.2.1.1	Small end and big end	66
4.2.1.2	Shank	67
4.2.2	Crank Shaft	68
4.2.2.1	Crank big end pin	69
4.2.2.2	Crank bearing pin	69
4.2.3	Crankshaft flange	69
4.2.4	Gears	70
4.2.5	Liner	71
4.2.6	Piston	71
4.2.7	Pulley	71
4.2.8	Bearing selection	72
4.2.9	Motor selection	72
4.2.10	Belt selection	72

<b>4.3</b>	<b>Measuring instruments</b>	<b>75</b>
	<b>4.3.1</b> Dial bore gauge	75
	<b>4.3.2</b> Fernier	76
	<b>4.3.3</b> Digital balance	76
<b>4.4</b>	<b>Test procedures</b>	<b>76</b>
 <b>Chapter five</b>		<b>80</b>
<b>Results and discussions</b>		<b>80</b>
<b>5.1</b>	<b>Introduction</b>	<b>81</b>
<b>5.2</b>	<b>Computer programs results</b>	<b>81</b>
	<b>5.2.1</b> Effect of engine with offset distance (d)	81
	<b>5.2.1.1</b> (intake-power) strokes	81
	<b>5.2.1.2</b> Piston displacement	84
	<b>5.2.1.3</b> Piston speed	85
	<b>5.2.1.4</b> Piston acceleration	85
	<b>5.2.1.5</b> Angle phi	86
	<b>5.2.2</b> Effect of (d), (r) and (b)	
	<b>5.2.2.1</b> (Intake – exhaust) stroke	86
	<b>5.2.2.2</b> Piston displacement	87
	<b>5.2.2.3</b> Piston speed	88
	<b>5.2.2.4</b> Piston acceleration	89
	<b>5.2.2.5</b> Angle (phi)	90
	<b>5.2.3</b> Motor driven engines	91
	<b>5.2.3.1</b> Inline crankshaft engine	91
	<b>5.2.3.2</b> Offset crankshaft engine	92

5.2.3.3	Twin crankshaft	92
5.2.3.4	Comparison between the three arrangements	93
5.2.3.4.1	Acceleration	93
5.2.3.4.2	Side thrust force	94
5.2.3.4.3	Force applied on piston	95
5.2.4	Combustion driven engine	95
5.2.4.1	Inline engine	95
5.2.4.2	Offset engine	96
5.2.4.3	Twin crankshaft	97
5.2.4.4	Comparison between the three arrangements	97
5.2.4.4.1	Torque	97
5.2.4.4.2	Side thrust force	98
5.3	Experimental Results	99
5.3.1	Wear characteristics	99
5.3.1.1	Inline engine	99
5.3.1.2	Offset crankshaft engine	110
5.3.1.3	Twin crankshaft engine	120
5.3.2	Comparison between the three arrangement	130
<b>Chapter six</b>		136
<b>Conclusions and suggestions for future work</b>		136
6.1	Conclusions	137
6.2	Suggestions for future work	138
<b>References</b>		139

<b>Appendices A</b>	FORTRAN program used for calculating piston displacement, speed, acceleration and the angle $\phi$ , with an offset crankshaft engine	143
<b>Appendices B</b>	FORTRAN program used to get the inline engine performance	143
<b>Appendices C</b>	FORTRAN program to calculate the offset crankshaft engine performance	144
<b>Appendices D</b>	FORTRAN program to calculate the twin crankshaft engine performance	146
<b>Appendices E</b>	Numerical solution of ordinary differential equations	147
<b>Appendices F</b>	Papers published during the course of present work	150