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A BRIEF REVIEW ON DYNAMIC ANALYSIS OF SLIDER-CRANK MECHANISM

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Abstract: The purpose of slider- crank mechanism is to convert reciprocating motion of a slider in to rotary motion of the crank or vice-versa. The application of slider-crank mechanism is in automobile engine by adding valve mechanism and it becomes a machine which converts the available energy (force on the piston) into desired energy(torque on the crank-shaft) . The torque is used to move a vehicle. Other applications include Reciprocating pumps, reciprocating compressors and steam engines derived from slider-crank mechanism. The slider-crank framework is a special 4-bar link configuration, which displays both linear and rotational motion at the same time. This method is frequently used for studying computer kinematics and the related dynamic forces in undergraduate engineering course. In analytical calculations, certain factors are often ignored that cause differences in results from experimental data. The objective of this MQP is to fabricate a working model of a slider crank mechanism that demonstrates the associated motion and provides means to measure kinematic properties, dynamic forces and cylinder pressure in various states of balance.

***Index Terms* - Linear Motion, Kinematic, Nonlinear controller, Rotary Motion, Slider-crank mechanism.**

I. INTRODUCTION

The “Standby” single acting, single piston pneumatic engine is one example of a slider crank mechanism. This engine utilizes a piston-wrist-pin-connecting rod construction that resembles an automotive engine. This type of engine provides a clear visual of the kinematics of a slider crank. The design can be poweredby compressed air or steam and is driven by pressure applied to one side of the piston within the cylinder. The single acting nature of this engine also more closely resembles the operation of an internal combustion engine. The design is simpler than alternative pneumatic and steam engines based on its lack of external eccentric valve gear. The design uses grooves on the crankshaft to port air into and out of the cylinder at specific times in its rotation. The Standby engine utilizes a valve system which is integrated into the crankshaft. This design characteristic was incorporated into the demonstration device.

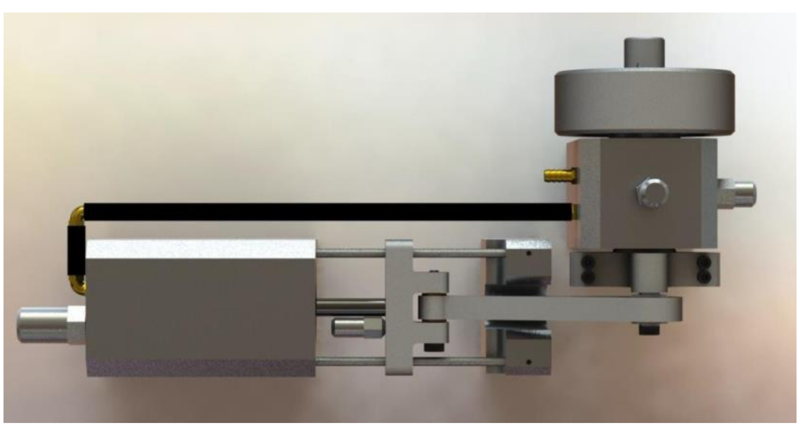


Fig 1: Single acting, Single Piston Pneumatic Engine

II. Analysis

(a) Kinematic Analysis

The kinematics of the slider-crank mechanism are evaluated at a rotational speed of 200rpm.

Angular Velocity $\eta := 200$ revolutions per minute

$$\omega := \frac{(2\pi\,\eta)}{60} \qquad \omega = 20.944 \text{ radians per second}$$

(b) Position Analysis

Slider Position

$$S(\theta) := L - \frac{R^2}{4 \cdot L} + R \cdot \left(\sin(\theta) + \frac{R}{4 \cdot L} \sin(2\theta) \right)$$

Crank Position

$$S_x(\theta) := R \cdot \sin(\theta)$$

$$S_y(\theta) := R \cdot \cos(\theta)$$

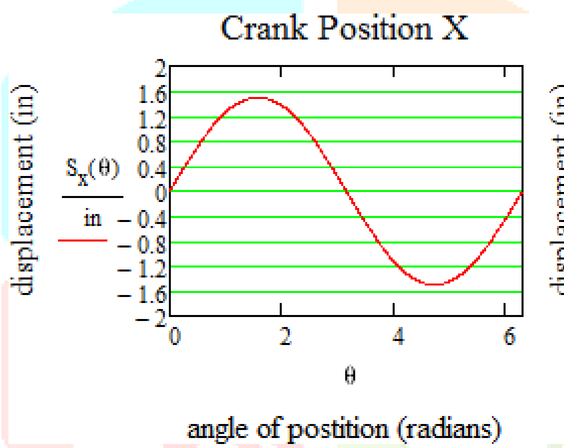


Fig 2: Crank X Displacement vs. Crank Angle

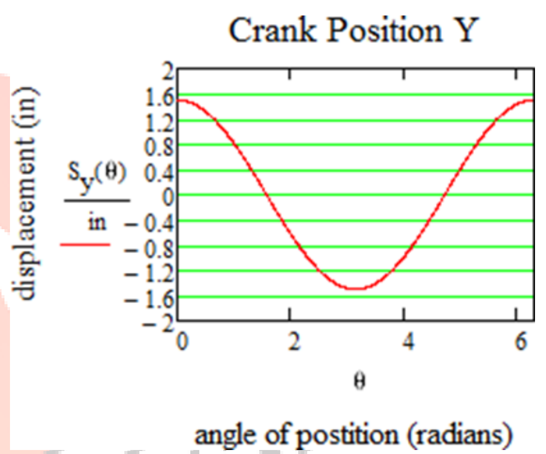
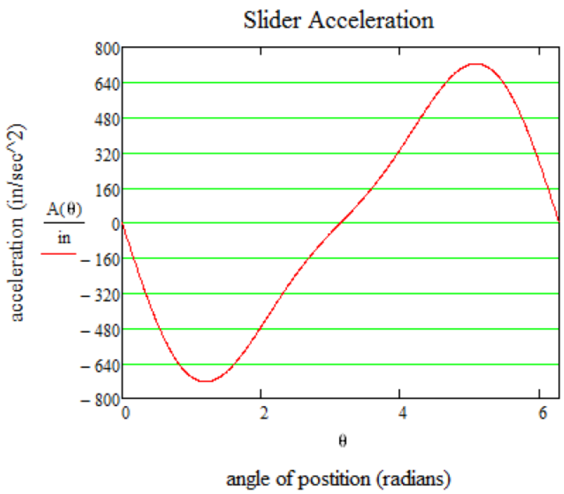


Fig 3: Crank Y Displacement vs. Crank Angle

(c) Acceleration Analysis

Slider Acceleration

$$A(\theta) := \left[-R\cdot\omega^2\left(\sin(\theta) + \frac{R}{L}\sin(2\theta)\right)\right]$$



Crank Acceleration

$$A_x(\theta) := -(\omega)^2\cdot R\cdot\sin(\theta)$$

$$A_y(\theta) := -(\omega)^2\cdot R\cdot\cos(\theta)$$

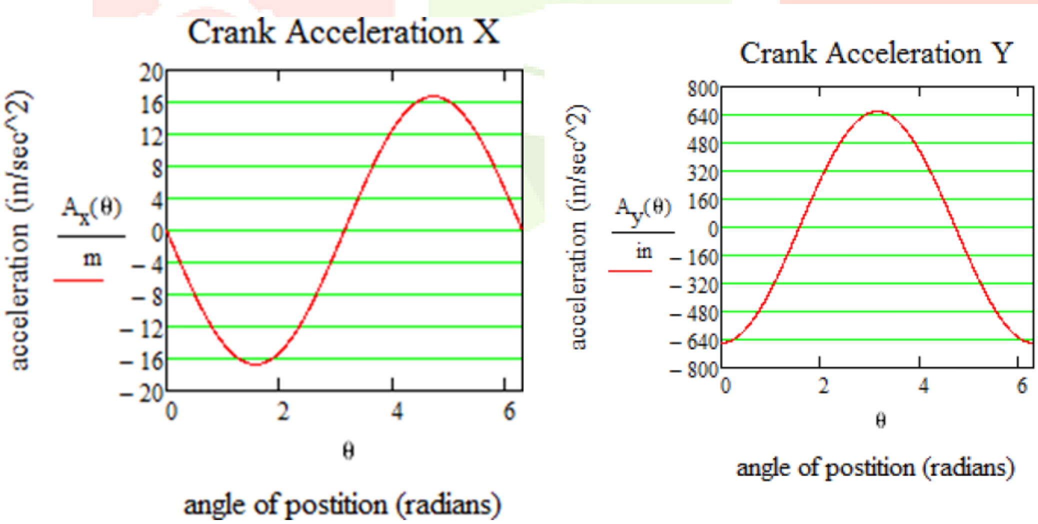


Fig 3: Y-Component of Crank Acceleration vs. Crank Angle

(d) Dynamic Force Analysis

Shaking Forces

Shaking force is defined as the sum of all forces acting on the ground plane of the system.

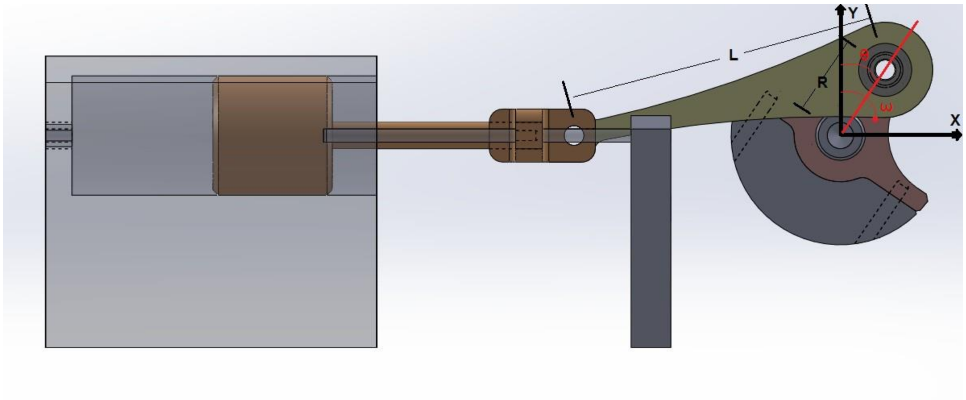
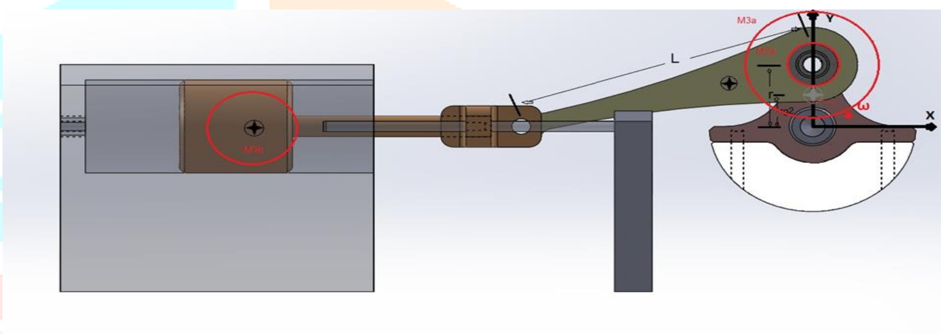


Fig 4: Slider-Crank Coordinate System and Variables

Unbalanced Configuration



M2a has a mass*radius product equal to that of the unbalanced crank M3a is equal to 2/3 the mass of the connecting rod
M3b is equal to 1/3 the mass of the connecting rod Effective mass at crank (lumped mass A)

Ma=Ma

Effective mass at slider (lumped mass B) Mb=M3b+(slider mass ?

$$F_{sx}(\theta) := -M_a \cdot \left(R \cdot \omega^2 \cdot \cos(\theta)\right) - M_b \cdot \left[R \cdot \omega^2 \left(\cos(\theta) + \frac{R}{L} \cdot \cos(2\theta)\right)\right]$$

$$F_{sy}(\theta) := -M_a \cdot \left(R \cdot \omega^2 \sin(\theta)\right)$$

Lumped Mass A $M_a := 0.467\text{kg}$

Lumped Mass B $M_b := 1.543\text{kg}$

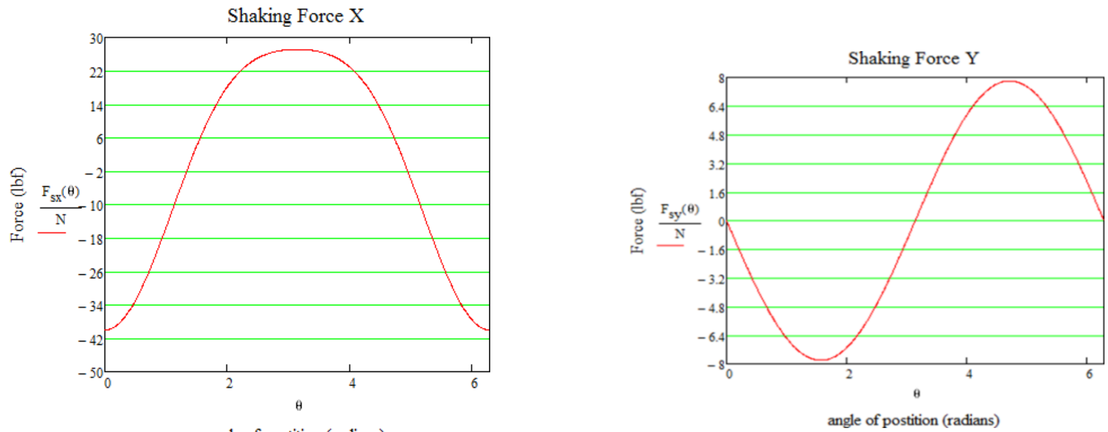
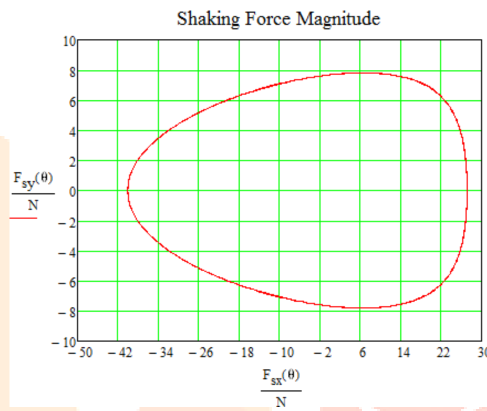
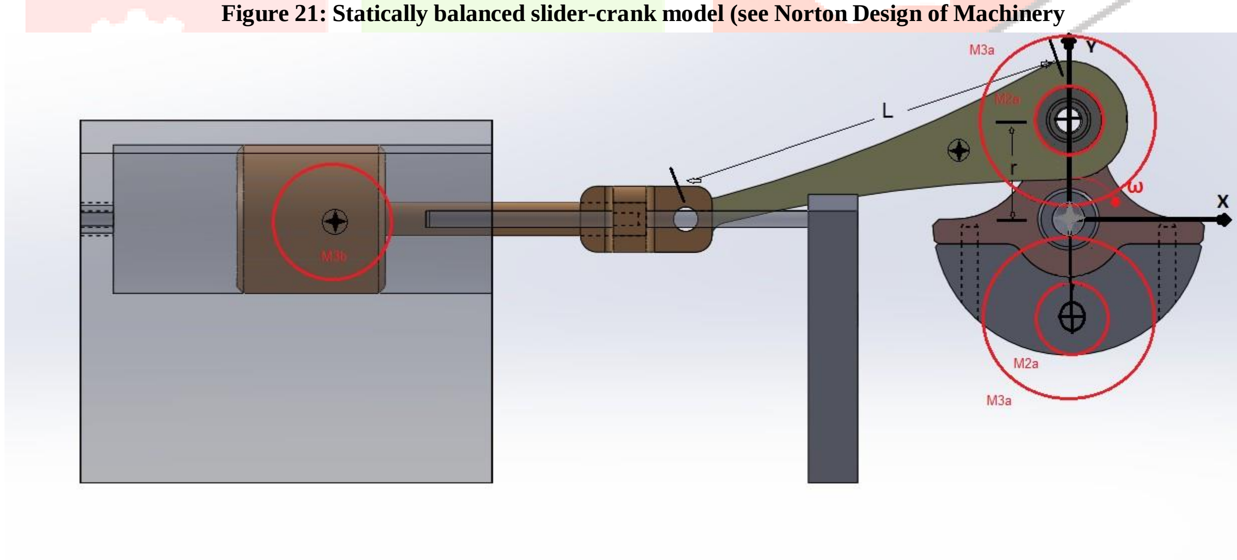


Figure 18: X-Component Shaking Force vs. Crank Angle (Unbalanced)



Statically Balanced Configuration

Figure 21: Statically balanced slider-crank model (see Norton Design of Machinery



M2a has a mass*radius product equal to that of the unbalanced crankM3a is equal to 2/3 the mass of the connecting rod

M3b is equal to 1/3 the mass of the connecting rod

Ma= M3a+M2a
Ma'= M3a'+M2a'
Effective mass at crank (lumped mass A)Ma=Ma-Ma'=0

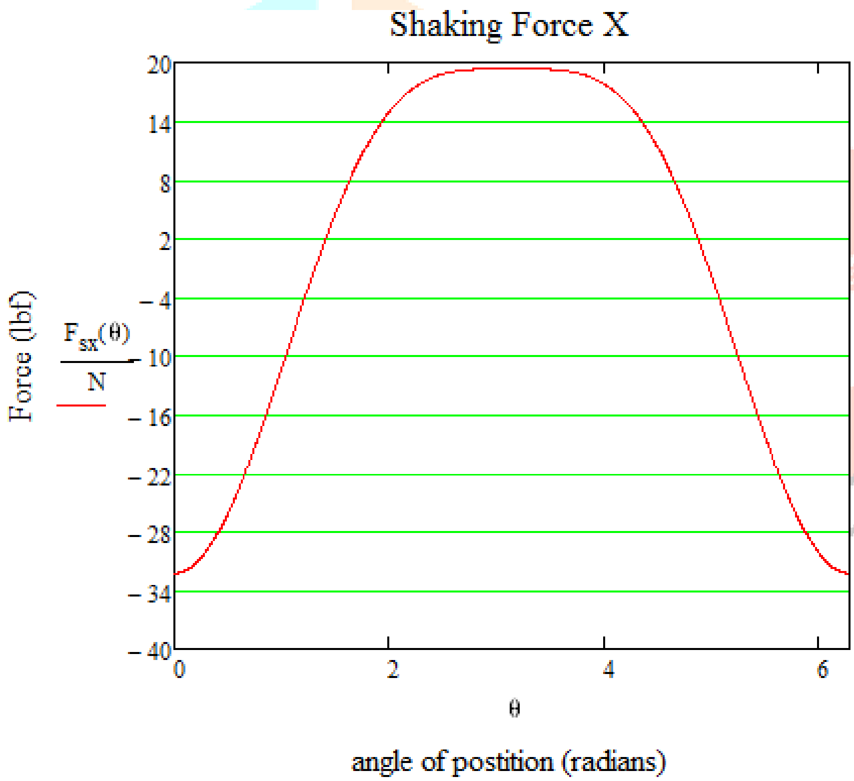
Effective mass at slider (lumped mass B) Mb=M3b+(slider mass)

$$F_{sx}(\theta) := -M_a\cdot\left(R\cdot\omega^2\cdot\cos(\theta)\right) - M_b\cdot\left[R\cdot\omega^2\left(\cos(\theta) + \frac{R}{L}\cdot\cos(2\theta)\right)\right]$$

$$F_{sy}(\theta) := -M_a\cdot\left(R\cdot\omega^2\sin(\theta)\right)$$

Lumped Mass A $M_a := 0\text{kg}$

Lumped Mass B $M_b := 1.543\text{kg}$



M2a has a mass*radius product equal to that of the unbalanced crankM3a is equal to 2/3 the mass of the connecting rod

M3b is equal to 1/3 the mass of the connecting rodMa= M3a+M2a

Ma’= M3a’+M2a’

1/2M_b≤**M_p**≤2/3M_b (the exact value depends upon the optimal operating speed) Effective mass at crank (lumped mass A)

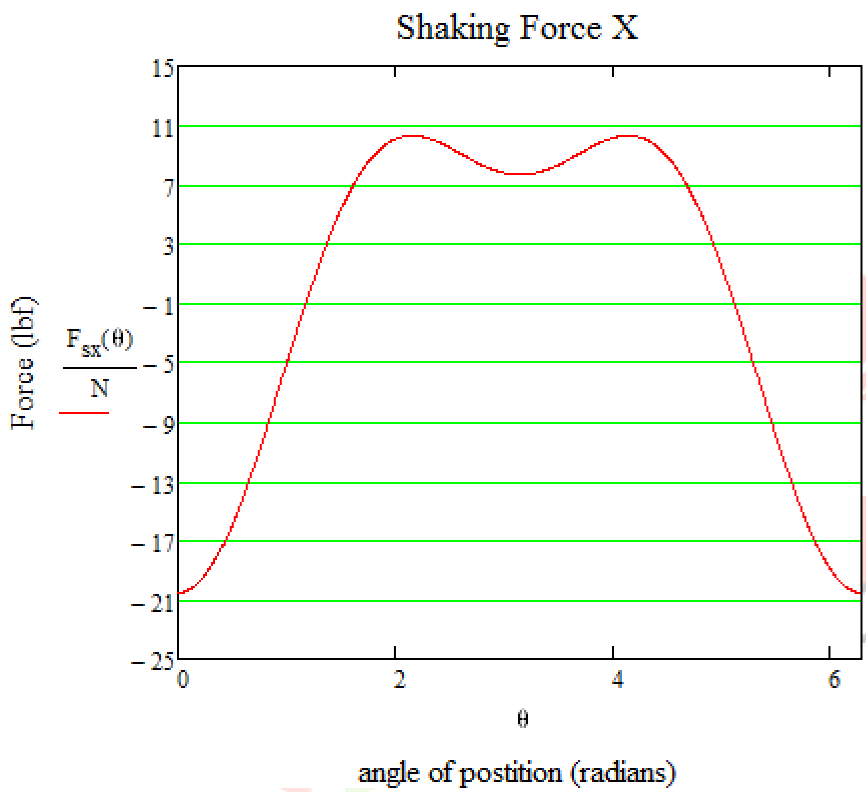
M_a=Ma Ma’-M_p=-M_p

Effective mass at slider (lumped mass B)

Lumped Mass A $M_a := -.7\text{kg}$

Lumped Mass B $M_b := 1.543\text{kg}$

M_b=M3b+(slider mass)



$$F_{sx}(\theta) := -M_a\cdot\left(R\cdot\omega^2\cdot\cos(\theta)\right) - M_b\cdot\left[R\cdot\omega^2\left(\cos(\theta) + \frac{R}{L}\cdot\cos(2\theta)\right)\right]$$

$$F_{sy}(\theta) := -M_a\cdot\left(R\cdot\omega^2\sin(\theta)\right)$$

$$A(\theta) := \left[-R\cdot\omega^2\left(\sin(\theta) + \frac{R}{L}\sin(2\theta)\right)\right]$$

The primary shaking force calculated previously using the lump mass model is selected as the magnitudeof force applied to the crankshaft through the connecting rod pin connection

Shaking force = 185 N

$$F_{sx}(\theta) := -M_a\cdot\left(R\cdot\omega^2\cdot\cos(\theta)\right) - M_b\cdot\left[R\cdot\omega^2\left(\cos(\theta) + \frac{R}{L}\cdot\cos(2\theta)\right)\right]$$

CONCLUSION

The production of the slider-crank mechanism was successful, with minimal variation from the original design required. The engine is capable of operating at angular velocities ranging from 80 to 330 rpm, using a balancing weight optimized for 200rpm. A wider range or operational speeds could be achieved with alternative balancing weights. The reduction in shaking force achieved through use of the balanceweights is apparent both visually and in recorded data.

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