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RESEARCH ARTICLE

EFFECT OF ROD STIFFNESS CHANGE ON THE PERFORMANCE OF RECIPROCATING PUMPS AND ITS STARTING REQUIREMENTS

[1,*]Ahmed M. Osman, [2]Mohamed A. El-Samanoudy and [2]Ashraf Ghorab,

[1]Faculty of Engineering, Ain Shams University, Egypt; Regional Manager, Odessa Separator Inc., USA
[2]Mechanical Power Engineering Department, Ain-Shams University, Cairo, Egypt

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ABSTRACT

Lots of solutions and methods were developed to deal with reciprocating pump systems' starting requirements either the reciprocating pump driven by a wind turbine or electric motor (Ford *et al.,* 2013). Decrease reciprocating pump-starting requirements in term of starting torque and starting current is a critical issue and should be considered in designing, sizing, and selecting reciprocating pump systems (Mahmoud Mohamed , 2013). In the present paper, the effect of rod stiffness change on the performance of reciprocating pumps and its starting requirements were experimentally studied using manufactured test rig equipped with closed loop reciprocating pump system. The system was driven by a squirrel cage electric motor through mechanical Scotch Yoke mechanism to transform rotational motion to linear (reciprocating) motion. Several experiments and the combination of the input parameters have been performed. Moreover, the outcomes and results obtained were explained, analyzed, discussed and the conclusion was drawn. The change of the reciprocating pump rod stiffness through the pneumatic cylinder improved the system efficiency and minimized the power consumption and the required starting torque. This approach experimentally proofed that the rod stiffness change has a great influence on the reciprocating pump performance and can decrease the reciprocating pump starting requirements. The starting torque requirements were decreased by an average of 19 %, power consumption corresponding to the peak starting torque were decreased by an average of 9% and the overall system performance and efficiency were increased by an average of 10%.

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INTRODUCTION

When describing reciprocating pump, we need first to highlight that it belongs to positive displacement pump family which pushes the amount of fluid toward in patches (Yohpe, 1968). The fluid enters the pump from the foot valve they pushed by a piston in patches by the moving valve and by the effect of the pump piston movement. When the suction stroke begins the piston opens the suction valve due to vacuum which resultsin suction while it's moving upward allows the fluid to enter the pump (Dhindsa, 1998). Whenthe discharge stroke starts, the piston close the suction valve and open the delivery valve by buildingup the pressure to deliver the fluid. This pump can be single, double or triple acting pistons. The material of the piston usually from brass, the foot valve material usually from metal and the piston was surrounded by cup-washer seals which usually manufactured from leather or polymers.

***Corresponding author: Ahmed M. Osman,**
Faculty of Engineering, Ain Shams University, Egypt; Regional Manager, Odessa Separator Inc., USA.
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Normally, the reciprocating pumps were the best candidate when high heads and low flows were needed beside its advantage of being a self-priming pump. The mandatory torque needed to start up the reciprocating pump or wind turbine usually about three times the normally required torque for normal operation. This high starting torque was representing high cost that should be paid to perform these requirements (Tackett *et al.,* 2008). High starting torque and rush current cause lots of problems and failures such as high-power loss in the transmission line, cable insulation's failure, and rotor heat up and may fail due to insulation failure. Several starting methods were used to start the reciprocating pumps such as: Direct on Line method, Star – Delta method, Variable Frequency (Speed) Drive, and Soft Starter. Direct online was one of the popular and simple methods usually used in starting reciprocating pumps' motors, but it can harm the system because it was associated with the highest starting current. In most cases, the starting current can be up to eight times the normal current. Direct on-line starting method was associated with high starting torque too in most cases, which can harm and damage the moving parts such as belts, coupling and lead to broken shaft cases (Gerin, 2007). Star – Delta was another way to start the reciprocating pump's motor, but it was used with no load applications or low starting torque. This way was used to decrease starting current especially with the induction motors. Recently, a new and efficient method of starting reciprocating pump's motor was developed called Variable Speed (frequency) Drive. This way starts the system smoothly from low speed to high speed and the drive can be adjusted to trip if the torque exceeds a certain percentage of the required torque (Frank, 1984). Soft starter was also a good starting method and was used to minimize the starting current of the reciprocating pump's motor, starting torque, which will lead to increase system run life and reduce system-needed maintenance. Mahmoud El-Hagar studied the effect of lots of parameters on the starting torque of reciprocating piston pump such as changing the Rotor and piston diameter, the speed of the wind, flow rate. This study was not limited to study the effect of these parameters on the starting torque, but he extends his study to find a way to reduce the starting torques of reciprocating piston pump by making a hole inside the piston. The main effect of this small hole was reducing the starting torque by reducing the pressure on the piston because all the fluid can find a path through this small hole which leads to decrease the pressure then the torque (Mahmoud Mohamed, 2013). He proofed that his way was one of the best ways to decrease the starting requirement of the reciprocating pump and wind turbine system. The piston hole improvement has a negative effect on the flow rate, but it has a lot of advantage on the pump performance especially on pump starting torque.

Gary L. Johnson discussed the power output from an ideal wind turbine, and then he compares it with the power output from a practical wind turbine. He did this study on different cases by changing the wind turbine driven shaft in each case (Johnson and Gary, 2001). He used different high-speed shafts with dissimilar materials to compare the output power from a wind turbine. This study proofed that changing the shaft material can change the overall system performance and its starting requirements. The shafts used in the experiment was stiff steel shaft and elastomeric shaft at different rotor angle to prove that the output power from the wind tubing increase by increasing the shaft stiffness. Steve Sabine developed a mathematical model using vector calculations and concept to proof that the stiffness has a pronounced effect on the machine behaviors and rotor parameters such as vibration and balancing. These changes in machine behavior were a result of changes in the stiffness (Sabin, 2000). Oil and Gas (2008) developed a new technique called "back-pressurized throttle bushing" to solve pump vibration problems by changing the stiffness of the rotor support. This technique was applied to several pumps to reduce the vibration and increase pump efficiency. GE is forcing some fluid from the discharge to back again to the bushing to change its stiffness and to develop its supporting force (Gas, 2008). By simulating different cases, it can be proven that in many cases vibration levels can be decreased with this solution. From all previous researches, the influence of rod stiffness change on the performance of machines, wind turbines and reciprocating pumps performance and its starting requirements can be recognized. The concept of changing stiffness in several ways as one of the effective factors which can enhance the performance and decrease the reciprocating pump starting requirements were studied in the research. In the present paper, a method was presented to decrease a reciprocating pump system starting requirements. The method, which will be followed, was to change the reciprocating pump rod stiffness by adding a variable pressure pneumatic cylinder and examine its effect on required reciprocating pump-starting torque when changing air pressure inside the pneumatic cylinder.

MATERIALS AND METHODS

Design of the experiment: To run the experiments the process wasdone in several phases: planning phase by studying all possible alternatives, and put guidelines for the equipment, parameters selections. Next phase was the design phase by making design, calculations, and modeling for a different part of the system such as a reciprocating pump, sizing motor, the design of Scotch Yoke mechanism, pneumatic cylinder modeling and load cell rated torque. The third phase was the implementation phase in which the designed test rig was manufactured, set up and planned experimentswere carried out. Then the final phase by analyses and compares the results so that valid and sound conclusion can be obtained, analyzed and compared.

Experimental setup: A test rig was manufactured to experimentally investigate the impact of rod stiffness change on the performance of reciprocating pumps and its starting requirements. The test rig key components were, test rig frame, reciprocating pump system, AC electric motor with its gearbox, Scotch Yoke mechanism, pneumatic system, instrumentations and measuring devices. The main assembly consists of three main systems: frame and fixation system, reciprocating pump system with closed loop suction and discharge, motor and gearbox system, Scotch yoke mechanism, pneumatic system, instrumentation, and drive. Figure 1 shows an isometric drawing of the overall assembly and Table 1 shows the overall assembly parts.

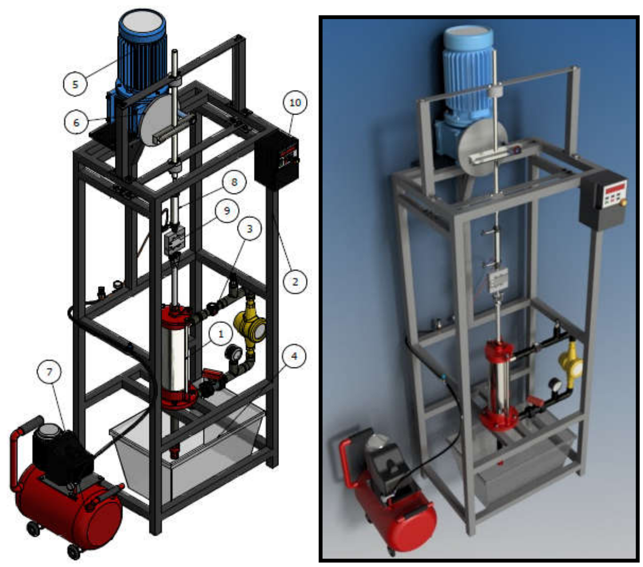

Figure 1. Test Rig Overview

Table 1. The Overall Assembly Parts

No.	QTY	DESCRIPTION	MATERIAL
1	1	Reciprocating Pumps	Stainless Steel
2	1	Frame and Fixation System	Steel
3	1	Closed Loop Suction and Discharge	Generic
4	1	Tank	Plastic
5	1	Motor and Gear Box System	Steel
6	1	Scotch Yoke Mechanism	Steel
7	1	Pneumatic System Air Compressor	Steel
8	1	Pneumatic System Double Acting Cylinder	Steel
9	1	Load Cell Transducer (S Shape)	Stainless Steel
10	1	Variable Speed (Frequency) Drive	Plastic

The objective of the fixing system was to fix the reciprocating pump system with closed loop suction and discharge in addition to the motor and gearbox system, mechanism, and all experiment parts. The reciprocating pump system including closed loop suction and discharge was installed at the bottom of the frame body, the motor, and its gearbox was based at the upper base welded to the frame body. The Scotch Yoke mechanism was placed in an intermediate position in between the reciprocating pump system and the motor system. The mechanism was sliding inside two axial bearings which weremounted in two bearing housing. The two axial bearings provide a proper guidance to the mechanism. Each bearing housing was attached to the frame through two steel supporters. The bearing housing carries the weight of the bearing and transfers it to the frame body through the two supporters, and the supporters were fixed to the frame by bolts and nuts. Pneumatic system gauges, regulator, and measurement devices such as AVO, Variable Speed (frequency) Drive, and Arduino were connected to the frame body. The reciprocating pump system consists of stainless-steel piston pump and a closed loop water circulation piping system. The reciprocating pump suction comes from the tank through the closed loop circulation piping system start then discharges to the same tank. The closed loop circulation piping system was equipped with an analog flow meter with 10% FS accuracy, digital flow meter, analog pressure gauge, digital pressure transducer and a valve. This reciprocating pump can operate with water negative level from 20 ft. to 25 ft.

To measure the discharge pressure of the reciprocating pump system, a pressure transducer with 1.5% FS accuracy was installed into the delivery pipe and the reading was monitored on a digital AVO. The discharge pipeline ended with a multi-position control valve. This control valve purpose was to have at least eight points of the flow rate at each experiment when measuring the reciprocating pump performance. The test rig was equipped with a tank to complete the closed loop system. The suction pipe takes the water from the tank and the delivery pipe was discharging to the same tank. Tank dimensions were 300 mm x 300 mm x 300 mm. The motor used in the experiment was a low voltage Single phase induction motor to provide the power to operate the reciprocating pump system. The motor power was 0.25 kW (0.33 HP), 0.84 power factor. Motor input parameters were 220-volt, 50 Hz. and the motor protection level was IP 55.The motor was equipped with gear reducer to decrease the motor maximum speed from 1390 revolution per minute to 60 revolutions per minute. Scotch Yoke mechanism was selected, designed, and manufactured to connect the reciprocating pump system and AC electric motor with its gearbox. This mechanism was selected because it was the simplest mechanism and used in converting rotational motion to linear (reciprocating) motion. A pneumatic system was manufactured to supply, regulate pressurized air to the pneumatic cylinder. Pressure gauge, pressure transducer and pressure regulator were installed to control and monitor the pressure of the air supplied by the compressor to the pneumatic cylinder. The change in the air pressure supplied to the pneumatic cylinder can be considered as a change in the reciprocating pump rod stiffness. Figure 2 shows Scotch Yoke Mechanism and pneumatic system.

DESIGN EQUATIONS AND CALCULATIONS

The pump output discharge (flow rate) (Q): To determine the pump size and capacity, the pump output discharge (flow rate) (Q)was calculated.

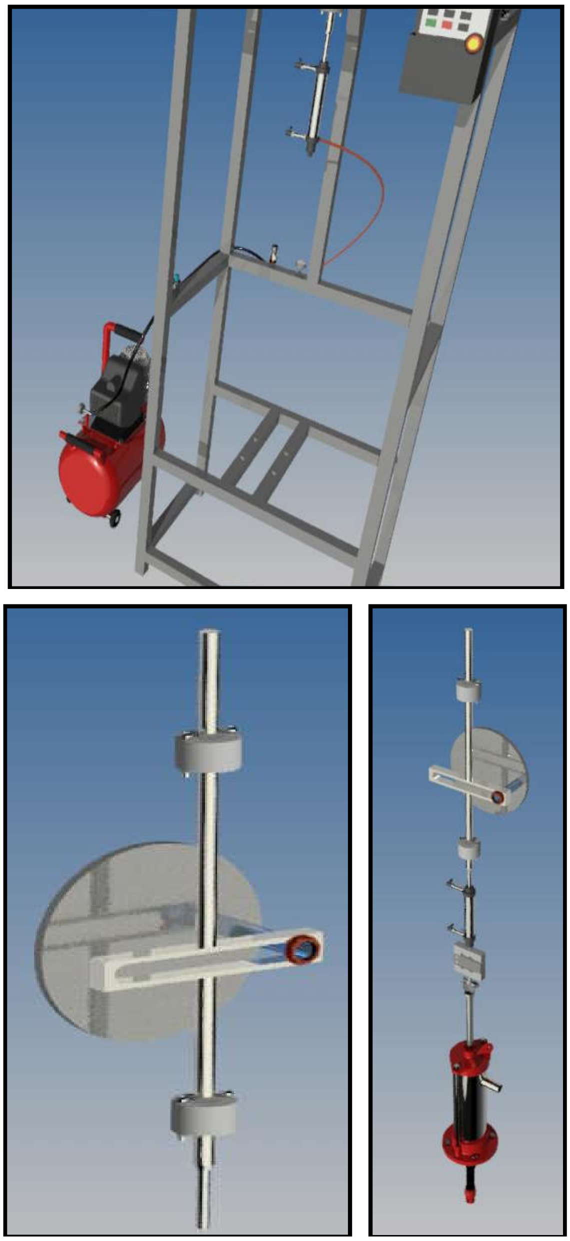

Figure 2. Scotch Yoke Mechanism and Pneumatic System

In a reciprocating pump, the pump output discharge (flow rate) (Q) was a function of piston diameter (D), stroke or length of piston travel (S), and a number of strokes per minute (N)[6]. The pump output discharge flow rate (Q)was calculated from equation (1).

$$Q = \frac{\pi}{4} D^2 \, S \, \frac{N}{60} \tag{1}$$

The maximum pump output discharge (flow rate) (Q) was calculated as:

$$Q = \frac{\pi}{4}(0.0762 \text{ m.})^2 (0.16 \text{ m.}) \frac{60 \text{ (rpm)}}{60} = 0.00073 \ m^3/s$$

The maximum pump output discharge (flow rate) (Q) 0.00073 m^3/s and by other units equal to 0.73 L/s.

Maximum operating pump pressure (P_{MAX}): Maximum operating pump pressure (P_{MAX}) that the pump can afford was a function of the head. The head (H). The maximum operating pump pressure (P_{MAX})was calculated from equation (2) as 0.43 was conversion factor from pressure (psi) to head(ft.)

$$P_{MAX}(\text{psi}) = \text{H (ft.)} \left(0.43 \frac{\text{psi}}{\text{ft.}}\right) \tag{2}$$

The maximum operating pump pressure (P_{MAX}) was calculated as:

$$P_{MAX}(\text{psi}) = 40 \text{ (ft.)} \left(0.43 \frac{\text{psi}}{\text{ft.}}\right) = 12.19\ psi$$

Motor brake horsepower (BHP): Motor brake horsepower (BHP) was a function of flow rate, pressure, gravity, fluid density and pump efficiency. Motor brake horsepower (BHP) was calculated from equation (3).

$$\text{Motor (BHP)} = \frac{Q\ P\ \rho\ g}{745\ X\ Efficiency} \tag{3}$$

The maximum motor break horsepower (BHP) required was calculated as:

$$\text{Motor (BHP)} = \frac{0.00073\ (12.19)(1000)\ (9.8)}{745\ \ (0.5)} = 0.23\ HP$$

The 0.29 HP motor was enough to meet the maximum power required.

DESIGN OF SCOTCH YOKE MECHANISM

The design of the Scotch Yoke mechanism was based on the maximum pump stroke that can be provided by the reciprocating pump. Figure 3 represents the Scotch Yoke mechanism which was attached to the electric motor shaft and connected to the reciprocating pump.

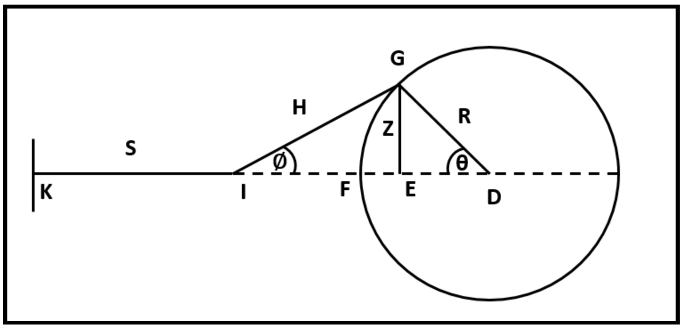

Figure 3. Scotch Yoke Mechanism Diagram

Let (R) be the radius of the crank, (H) connecting rod length, ($S_{CUTTING}$) was the length of the cutting stroke, Angle (θ) was EDG in the triangle EDG and Angle Φ was EIG. In triangle EGI, the Cos (Φ) was defined as: The length of the Cutting stroke (S) was calculated from equation (4).

$$X = DF + FK - EI - DI \tag{4}$$

$$\cos\emptyset = \frac{EI}{H}$$

$$\text{Then EI} = H\ \cos\emptyset \tag{5}$$

In Triangle DEG, the Cos θ was defined as

$$\cos\theta = \frac{DE}{R}$$

$$Then\ DE = R\ \cos\theta \tag{6}$$

$$\sin\emptyset = \frac{Z}{H} \tag{7}$$

$$\sin\theta = \frac{Z}{R} \tag{8}$$

$$\cos\emptyset = \sqrt{(1 - \sin^2\emptyset)} \tag{9}$$

From equation (7) and (8) in equation (9), equation (10) can be obtained.

$$\cos\emptyset = \sqrt{\left(1 - \frac{R^2\sin^2\theta}{H^2}\right)} \tag{10}$$

$$Assume\ n = \frac{connecting\ rod\ length}{Crank\ radius} = \frac{H}{R}$$

$$\cos\emptyset = \sqrt{1 - \frac{\sin^2\theta}{n^2}} \tag{11}$$

If n≫1, by binomial expansion then:

$$\cos\emptyset = 1 - \frac{\sin^2\theta}{2n^2} \tag{12}$$

By substitute from equation (5), (6) and (12) in equation (4), equation (13) can be obtained.

$$S_{CUTTING} = R + H - H\left[\left(1 - \frac{\sin^2\theta}{2\,n^2}\right)\right] - R\cos\theta \tag{13}$$

$$S_{CUTTING} = R\,(1 - \cos\theta) + H\left[1 - 1 + \left(1 - \frac{\cos 2\theta}{4\,n^2}\right)\right] \tag{14}$$

$$S_{CUTTING} = R\,(1 - \cos\theta) + \frac{R^2}{4\,H}(1 - \cos 2\,\theta) \tag{15}$$

The pump stroke (S) required was 160 mm. as per the pump manufacturer and it was a function of the radius of the crank (R), connecting rod length(H) and the cutting stroke ($S_{CUTTING}$).

$$\text{Pump Strock (s)} = R + H + S_{CUTTING} \tag{16}$$

At maximum stroke (minimum cutting stroke) at Ø = 0, θ = 0
$\text{Pump Strock (s)}_{max.} = 160\text{mm} = R + H + 0$

Then the in the radius of the crank (R) and connecting rod length (H) can be 80 mm.

Experiment limitations

The motor speed with gearbox system was sixty revolutions per minute, by adding the Variable Speed (frequency) Drive, the speed can be changed from zero to sixty revolutions per minute by using the speed regulation function which was built in the Variable Speed (frequency) Drive. Although the complete electric system capability (motor plus gearbox plus Variable Speed (frequency) Drive) could deliver up to sixty revolutions per minute, the maximum operating revolutions per minute that could be handled in the experiment was twenty-three revolutions per minute. The reason for this limitations was not exceeding the sensors and transducers' rating in addition to secure the system vibration and stability because after twenty-three revolutions per minute the system construction suffers from high vibration and stability issue. The lower speed also helps to enhance the system overall run life and reduce the sealing wear. The load cell transducer measures the force acting on the reciprocating pump rod during the suction stage and discharge stage. The load cell transducer limitation (rated reading) was five volts which exceed when we operate the experiment above twenty-three revolutions per minute. The Pressure transducer measures the discharge pressure at reciprocating pump discharge. The Pressure transducer limitation (rated reading) was 24 Bar which can cover the experiment all inputs combinations. The digital flow meter measures the flow rate at the reciprocating pump discharge. The digital flowmeter (rated reading) was 2-60 Liter per minute which was equal up to 3.6-meter cube per hour. This range can cover the experiment need as the maximum flow rate in the experiment did not exceed 0.5-meter cube per hour. The compressor capacity in the local market was eight bar supplying pressure with twenty-liter capacity. The pressure needed to start the experiment and to overcome the Scotch Yoke mechanism weight was six Bar. The compressor can handle up to eight bar which was sufficient for the experiment needs.

Input data and input combination

Table 2 presents the input parameters and the experiment measurable and calculated outputs.

Table 2. Input (Design Parameters) and the Experiment Output

Inputs	Measurable Output	Calculated Output
Motor Speed	Electric Power Consumed	Mechanical Power
Pneumatic Cylinder Air Pressure	Force Acting on Reciprocating Pump Rod	Efficiency
	Reciprocating Pump Discharge Pressure	Force Acting on Reciprocating Pump Rod
	Reciprocating Pump Flow Rate	

Input combinations result from varying different input parameters such as motor speed and air pressures inside the pneumatic cylinder. A big number of experiments were performed with changing the previously mentioned parameters. The experiments were categorized into six sets, each set will contain six experiment which will resultinthirty-six experiment. In each set, the air pressures inside the pneumatic cylinder will be constant and the motor speed will be changed. Table 3 presents the combination done of each input parameters.

RESULTS AND DISCUSSION

In the experiment, the input design parameters were motor speed and the pneumatic cylinder air pressure which reflect the change in the reciprocating pump rod stiffness. Table 4 shows the input parameters of the experiment. The reciprocating pump performance was measured by varying different design parameters such as, air pressures inside the pneumatic cylinder, which reflects the stiffness variation and motor speed. Then, the results were used to compare the performance achieved while changing the design parameters .Table 5 shows measurable and calculated output values of the sensors and the transducers at eight different points on the curve. Figure 4 shows the performance curve of the reciprocating pump (flow rate vs. head) for this input combination. Figure 5 and figure 6 show flow rate vs. torque and flow rate vs. efficiency for the same input combination. To give full overview about the effect of varying the input parameters, Figure 7 combines all results of varying pump shaft rotational speed (N) of10 RPM, 13 RPM, 15 RPM, 17 RPM, 20 RPM, and 23 RPM while increasing the air pressure inside the pneumatic cylinder which reflects the reciprocating pump rod stiffness change and shows its effect on the reciprocating pump maximum rod torque (starting torque). Figure 8 combines all results of varying pump shaft rotational speed (N) of10 RPM, 13 RPM, 15 RPM, 17 RPM, 20 RPM, and 23 RPM while increasing the air pressure inside the pneumatic cylinder which reflects the reciprocating pump rod stiffness change and shows its effect on maximum power consumption during starting torque. Figure 9 combines all results of varying pump shaft rotational speed (N) of 10 RPM, 13 RPM, 15 RPM, 17 RPM, 20 RPM, and 23 RPM while increasing the air pressure inside the pneumatic cylinder which reflects the reciprocating pump rod stiffness change and shows its effect reciprocating pump efficiency. The drive input factor in this experiment was the air pressure inside the pneumatic cylinder which reflects the reciprocating pump rod stiffness change. While increasing air pressure inside the pneumatic cylinder, monitor the various pump output parameters and starting requirements behavior were essential to evaluate the effect of the stiffness change on the overall pump performance. When starting the reciprocating pump, the starting conditions were monitored at an air pressure inside the pneumatic cylinder equal zero Bar. Then keep monitoring while increasing the air pressure inside the pneumatic cylinder to 7.8 Bar. When starting the reciprocating pump with initial air pressure equal Zero bar, the reciprocating

Table 3. Combination of Input Parameters

Set No.	Pneumatic Cylinder Air Pressure (BAR)	Motor Speed (RPM)	Set No.	Pneumatic Cylinder Air Pressure (BAR)	Motor Speed (RPM)
1st Set	ZERO	10	2nd Set	6.2	10
	ZERO	13		6.2	13
	ZERO	15		6.2	15
	ZERO	17		6.2	17
	ZERO	20		6.2	20
	ZERO	23		6.2	23
3rd Set	6.45	10	4th Set	7	10
	6.45	13		7	13
	6.45	15		7	15
	6.45	17		7	17
	6.45	20		7	20
	6.45	23		7	23
5th Set	7.4	10	6th Set	7.8	10
	7.4	13		7.8	13
	7.4	15		7.8	15
	7.4	17		7.8	17
	7.4	20		7.8	20
	7.4	23		7.8	23

Table 1. Input Parameters of the Experiment

Parameters	Input Value
Pneumatic Cylinder Air Pressure (BAR)	ZERO
Motor Speed (RPM)	10

Table 2. Measurable and Calculated Output Values of the Experiment

	Power (W)	Torque (Nm.)	Pump Discharge Pressure (m.)	Pump Flow Meter m^3 /h	Efficiency%
Point 1	122	31.8	2.07	0.18	27%
Point 2	120	27.2	1.03	0.20	24%
Point 3	116	27.2	0.26	0.20	25%
Point 4	115	25.9	0.26	0.21	24%
Point 5	114	25.9	0.26	0.21	24%
Point 6	113	25.9	0.26	0.21	24%
Point 7	112	25.9	0.26	0.21	24%
Point 8	110	24.6	0.26	0.22	23%

Table 3. Values of All Parameters While Starting the Reciprocating Pump with Initial Air Pressure

Air Pressure Inside Pneumatic Cylinder (Bar)	Motor Speed (RPM	Starting Torque(N.m.)	Power Consumption (W)	Efficiency%
	10	33	115	27 %
Zero	13	45	145	33 %
	15	48	171	35 %
	17	52	201	33 %
	20	58	240	38 %
	23	68	316	44 %

Table 4. Values of All Parameters While Starting the Reciprocating Pump with Maximum Air Pressure

Air Pressure Inside Pneumatic Cylinder (Bar)	Motor Speed (RPM	Starting Torque(N.m.)	Power Consumption (W)	Efficiency%
	10	27	99	30 %
7.8	13	31	131	35 %
	15	34	155	37 %
	17	45	180	39 %
	20	54	225	42 %
	23	57	287	48 %

Table 5. System Enhancement at Different Speed

Motor Speer (RPM)	Peak Starting Torque Decrease (%)	Power Consumption Decrease (%)	Overall System Efficiency Increase (%)
10	18 %	14 %	11 %
13	31 %	10 %	6 %
15	29 %	9 %	6 %
17	13 %	6 %	18 %
20	7 %	6 %	11 %
23	16 %	9 %	9 %

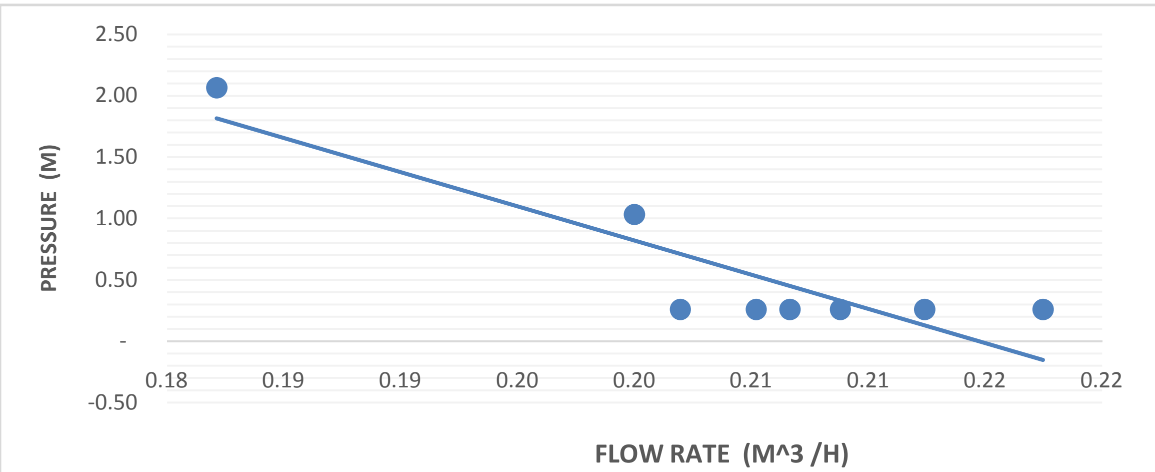

Figure 4. Performance Curve of the Reciprocating Pump (Flow Rate vs. Head)

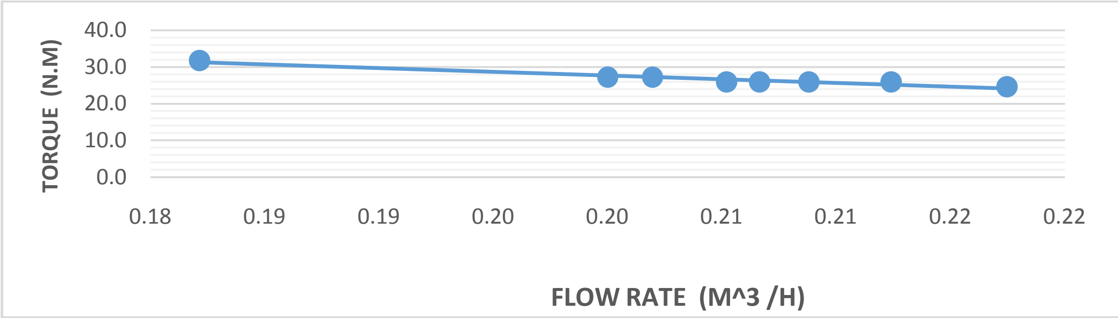

Figure 1. Flow Rate vs. Torque

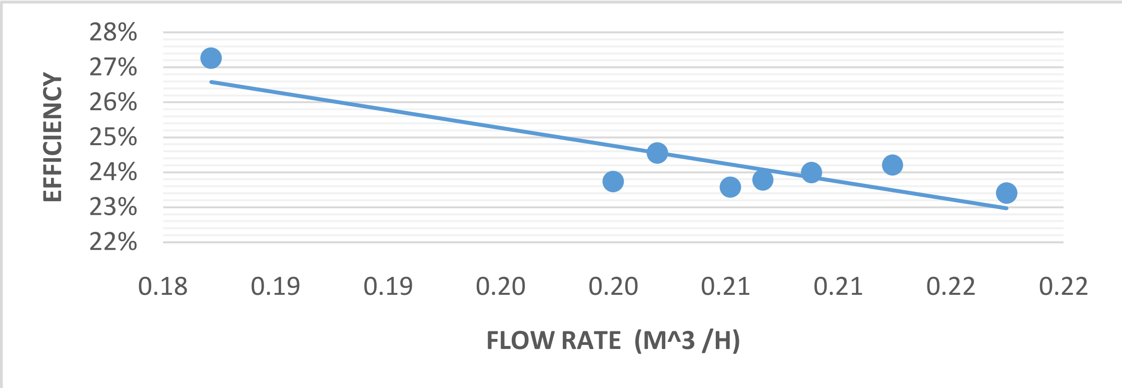

Figure 2. Flow Rate vs. Pump Efficiency

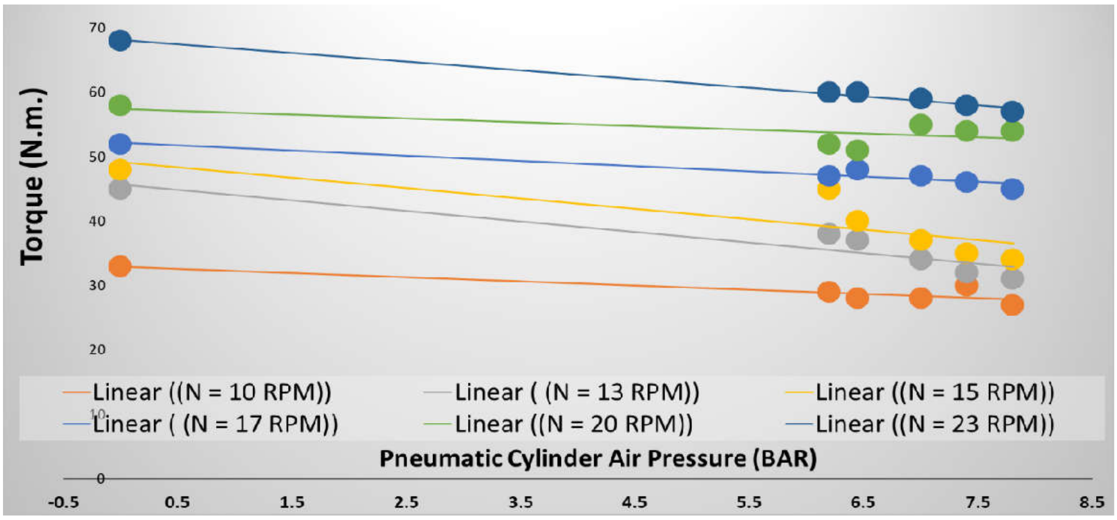

Figure 3. Performance Comparison While Changing Air Pressure and RPM against Torque

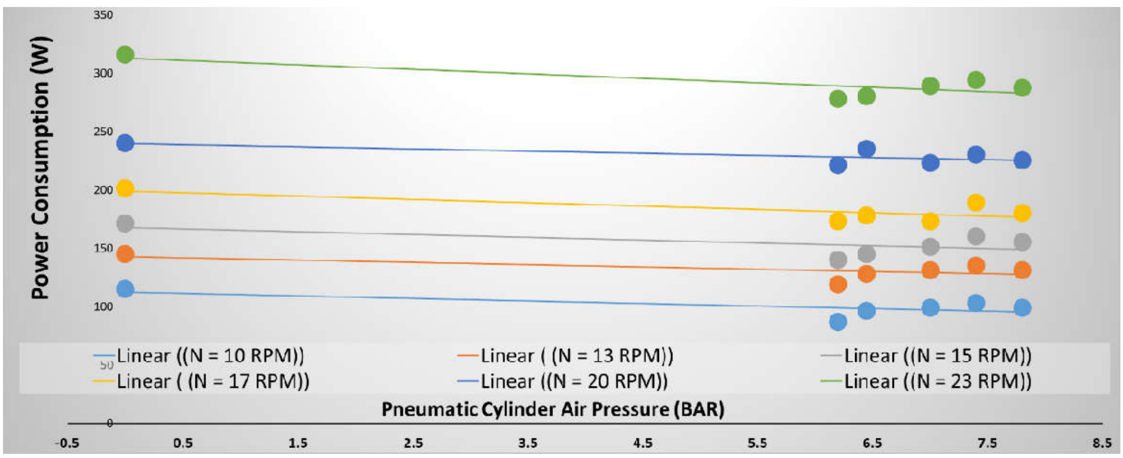

Figure 4. Performance Comparison While Changing Air Pressure and RPM against Power Consumption

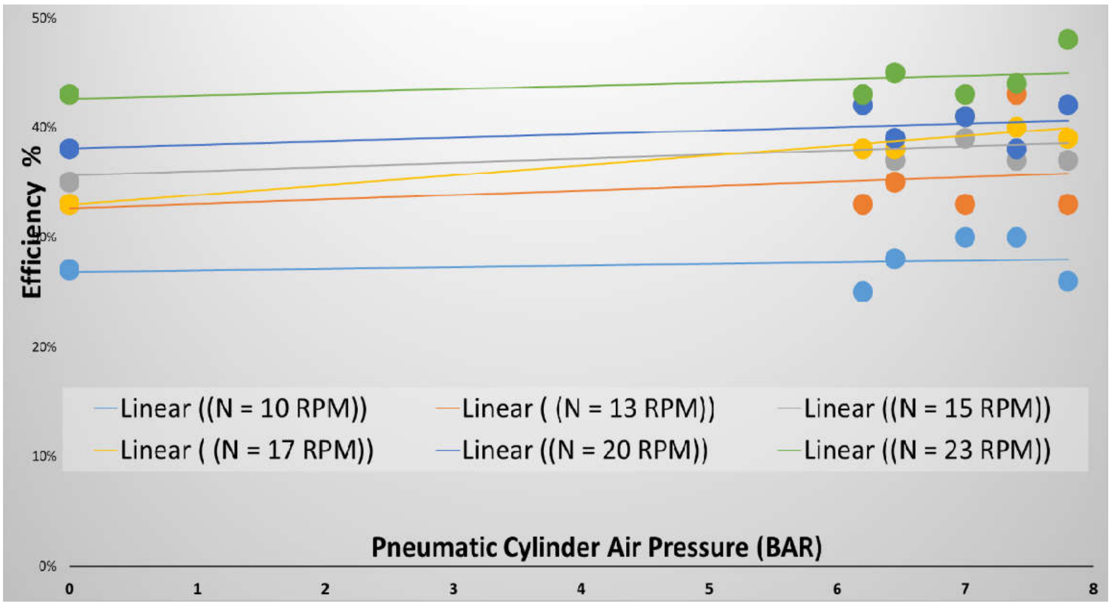

Figure 5. Performance Comparison While Changing Air Pressure and RPM against Efficiency

pump starting torque and the power consumption were at their maximum value while the reciprocating pump efficiency started with its minimum value. Table 6 shows values of the reciprocating pump starting torque, the power consumption, and pump efficiency while starting the reciprocating pump with initial air pressure equal Zero Bar. By increasing the air pressure inside the pneumatic cylinder from initial value (Zero Bar) through values 6.2 bar, 6.5 bar, 7 Bar, and 7.4 Bar, till its highest value 7.8 Bar.

The starting requirements started to decrease. After the air pressure inside the pneumatic cylinder reaches to 7.8 bar, the reciprocating pump starting torque and the power consumption were at their minimum value while the reciprocating pump efficiency increased to its maximum value.
Table 7 shows values of the reciprocating pump starting torque, the power consumption, and pump efficiency while starting the reciprocating pump with maximum air pressure equal 7.8 Bar. Increase air pressure inside the pneumatic cylinder which reflects the reciprocating pump stiffness change lead to decrease the reciprocating pump starting requirements such as starting torque and power consumption and increasethe overall system performance and efficiency.

Conclusion

An approach was presented to decrease a reciprocating pump system starting requirements such as peak starting torque as one of the main critical factors in reciprocating pump starting requirements. By following this approach the starting requirements such as peak starting torque and power consumption can be decreased, overall system performance and efficiency can be increased. Table 8 shows system enhancement at different speed. This approach experimentally proofed that the rod stiffness change has a great influence on the on reciprocating pump performance and can decrease the reciprocating pump starting requirements. From the results above,the starting torque requirements were decreased by an average of 19 %, power consumption corresponding to the peak starting torque were decreased by an average of 9% and the overall system performance and efficiency were increased by an average of 10%.

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