Scientific Research Publishing

International Journal of Modern Nonlinear Theory and Application, 2022, 11, 53-59
https://www.scirp.org/journal/ijmnta
ISSN Online: 2167-9487
ISSN Print: 2167-9479

An Experimental Determination of the Supply and Exhaust Pressures in an Electro-Pneumatic Clutch Actuation System

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How to cite this paper: Ndubuisi, P.-D.I., Eneh, I.I. and Nnaji, A.C. (2022) An Experimental Determination of the Supply and Exhaust Pressures in an Electro-Pneumatic Clutch Actuation System. *International Journal of Modern Nonlinear Theory and Application*, **11**, 53-59.
https://doi.org/10.4236/ijmnta.2022.113004

Received: July 1, 2022
Accepted: July 26, 2022
Published: July 29, 2022

Abstract

Inlet and outlet orifices in an actuation chamber are sources through which the supply and exhaust pressures pass during the actuation process in clutch systems. They are key ingredients in an actuation chamber and are very phenomenal in heavy-duty vehicle operation. It is these pressures that initiate linear or rotary motions in drive systems. The pressure actions are processed in an enclosure termed an actuation chamber. Oftentimes, the forces or pressures produced in an actuation chamber are unknown and immeasurable owing to a lack of precise instruments to accomplish them. This challenge can only be approached via an improvised technique that requires experimentation. This is precisely what this presentation is all about. The knowledge of these parameters is important in the study of the actuation process in electro-pneumatic clutch systems of heavy-duty vehicles. The study was done with a Mercedes Benz Actros Truck Model MP 2, 2031 Actuator chamber. An empirical and analytical approach was adopted. Meter rule, Venire Callipers and Mass Spring Balance were deployed for the experiments. Piston coil or spring, clutch distance in the actuator, the cross-sectional diameter of the actuator, and displacement in the free lengths of the coils among others were measured. The results of the experiments were analysed and used to determine the values of the supply (inlet) and exhaust (outlet) pressures which results stood at 9.61 bars and 11.299 bars, respectively.

Keywords

Actuation, Chamber, Electro-Pneumatic, Instruments, Pressure

1. Introduction

In automotive industries, the transmission process is a mechanism used in transferring the mechanical power of a motor vehicle from the engine to the wheels. Actuation on the other hand is important in any physical process that requires mechanical movement. The actuation process is a system that utilises its outputs to achieve a control action on a machine or device, with the ultimate aim of producing a linear or rotary motion. A device or element deployed for this purpose of actuation is an actuator [1]. Said in another way, an actuator is an element or device that transforms a mechanical movement from one form to the other such as linear or rotary motions. This transformation is usually required in any physical process. Both transmission and actuation processes are independent terms but are often inseparable in a drive line system. Driveline system components are those elements that transmit and control power and motion. They are the brakes and the clutches. A Brake system can be explained from the viewpoint of a clutch system in the sense that it can assist in reducing the motion of a moving body like a clutch. But unlike a clutch where both shafts are rotatory, one of the shafts of a brake is fixed or held stationary while the other shaft rotates just like in a clutch. The essence of clutching is to bring about the partial or total engagement of two shafts that are rotating on their own independently at different speeds. By so doing, it may result in a reduced speed of the shafts when partially engaged [2]. If both shafts are held together, it can result in a total stoppage of the rotation of the shaft. This is a total engagement of the clutch. These independently rotating shafts can be linked together by direct mechanical lockup or by mechanical friction. It can also be linked by either pneumatic or hydraulic or electromagnetic or a combination of either electro-pneumatic or electro-hydraulic forces [3]. The process that actualises this linear or rotary motion in drive systems is initiated in an enclosure termed an actuation chamber. A typical actuator chamber is shown pictorially in **Figure 1** below.

The forces or pressures produced in an actuation chamber are sometimes difficult to be measured due to the unavailability of precise instruments to accomplish the task. Faced with such a challenge necessitates an improvised technique that requires experimentation. This is exactly what is explored in this piece. The knowledge of these parameters is important in the study of the actuation process in electro-pneumatic clutch systems of heavy-duty vehicles. Thus, the paper demonstrates an alternative solution for solving critical clutch actuation needs in an

(a) (b)

Figure 1. Typical actuator chamber of a heavy duty vehicle. (a) Longitudinal view (b) Cross-sectional view.

electro-pneumatic transmission environment.

2. Methods

The research design method adopted was empirical and analytical in nature. Experiments were done with a Mercedes Benz Actros Truck model MP 2, 203 as follows.

1) Meter rule was used to measure free lengths of the piston coil or sprig and displacements in the free lengths of the coils as weights or loads of twenty grams are added incrementally.

2) Mass Spring Balance System was used to measure the piston spring compression with respect to applied force or weight. This weight or force approximates to the supplied pneumatic force required to fully compress the piston springs during clutch disengagement in an actuation process. The diminishing linear lengths of the spring are measured with a meter rule.

3) Venire Callipers were also used for the precise measurements of the diameter and thickness of the spring coil.

In this manner, data for clutch actuation parameters were obtained through measurement and tabulated in Table 1.

The pictorial views of the experimental set ups are shown in Figure 2 below.

3. Experimental Design, Analysis and Results

It is noteworthy that in an ideal situation, the supply and exhaust pressures in an

Table 1. Measured parameters.

S/N	Parameters	Readings
1	Piston coil diameter (D)	4.01 cm
2	Piston wire diameter (d)	0.235 cm
3	Piston coil free length (L_f)	17 cm
4	Number of Piston coils (N)	7
5	Compressor capacity	8 bars
6	Diameter of the inlet/supply port or valve (Vs)	1.41 cm
7	Diameter of the outlet/exhaust port or valve (Ve)	1.30 cm

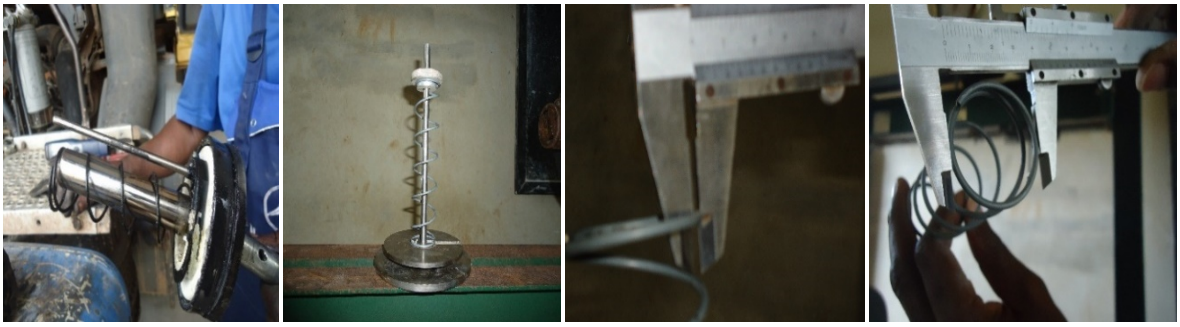

Figure 2. Pictorial views of the experimental set up.

electro-pneumatic clutch actuation system are not matters that should be south for. Rather, it should be readily available, however, this is not the case, hence, this presentation.

The data presented in Table 2 below, shows the result of compressing the piston spring under varying weights. The data is fundamental to the analysis that follows in arriving at the needed derived parameters.

The plot of Columns 1 and 4 is presented in the graph of Figure 3. The graph is also useful in the derivations that subsequently follows and which will ultimately lead to the required parameters.

Table 2. The mass/spring compression of piston coil.

Force/weight (N)	Free Length, L_f (cm)	Length Variations, L_i (cm)	L_f-L_i (cm)
0	17	17	0
5	17	16.5	0.5
10	17	15.9	1.1
15	17	15.5	1.5
20	17	14.9	2.1
25	17	14.5	2.5
30	17	14.0	3.0
35	17	13.5	3.5
40	17	12.9	4.1
45	17	12.5	4.5

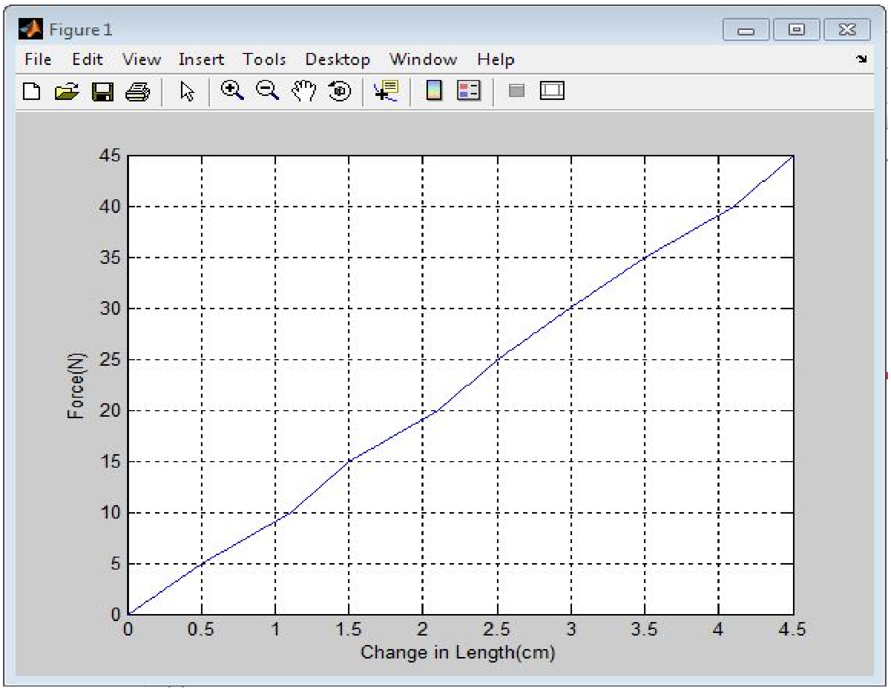

Figure 3. Graph of force/weight (N) and change in length {L_f-L_i} (cm).

[4] noted that

Solid length (L_s) of spring is defined by

$$L_s = d(N+1). \tag{1}$$

where N is the number of piston ring coils, solving

$$L_s = 0.235(7+1) = 1.88 \text{ cm}.$$

Maximum deflection (Δ_L) of the spring is given by

$$\Delta_L = L_f - L_s. \tag{2}$$

where L_f is for spring free length. Substituting, we have

$$\Delta_L = 17 - 1.88 = 15.12 \text{ cm}.$$

Compressing the piston spring beyond 12.5 cm as shown in **Table 2**, Column 3 may lead to the spring yielding, hence defeating the essence of the experiment. Thus, by applying the mechanics of spring materials standard analysis for springs under tensile or compressive stresses, [4] observed that

Spring rate (K) is defined as

$$K = \frac{\Delta W_1}{\Delta L_1} = \frac{\Delta W_2}{\Delta L_2}, \tag{3}$$

where ΔW = change in weight.and ΔL = change in length.

From the graph of **Figure 3**, $A = (x_1, y_1) = (3.5, 35)$ and $B = (x_2, y_2) = (2.2, 20)$. Points A and B are the specific X and Y coordinates in the graph of **Figure 3**.

$$\text{Slope of line } AB = k = \frac{\Delta W_1}{\Delta L_1} \tag{4}$$

$$\text{Slope of line } AB = \frac{35-20}{(3.5-2.2)\times 10^{-2}} = \frac{15}{1.3\times 10^{-2}} = 1.15385\times 10^{3} \text{ N/m}.$$

Maximum force to deflect and collapse the spring is $w_{1\max}$.

From Equation (3),

$$w_1 = k\Delta L_1. \tag{5}$$

By substitution, $k = 1.15385\times 10^{3}$ N/m and $\Delta L_1 = 1.3\times 10^{-2} = 0.013$ m

$$w_{1\max} = k\Delta L_s = 1.15385\times 10^{3} \text{ N/m}\times 0.013 \text{ m} = 0.015\times 10^{3} \text{ N}$$

Pressure at the inlet/supply valve or port is given by

$$P_s = \frac{\text{Force}}{\text{Area}} \tag{6}$$

$$P_s = \frac{w_{1\max}}{\text{Area of valve}}$$

$$P_s = \frac{0.015\times 10^{3} \text{ N}}{\frac{\Pi D^2}{4}} = \frac{0.015\times 10^{3} N \times 4\times 10000}{\Pi\times 1.41\times 1.41} = \frac{0.6\times 10^{7}}{6.2458}$$

$$= 0.961\times 10^{6} \text{ N/m}^2 = 9.61\times 10^{5} \text{ N/m}^2 = 9.61 \text{ bars}$$

However, the compressor capacity is only 8 bars. Hence the difference of 1.61 bars in the result above is assumed to come from the hydraulic source hence the pneumatic pressure is normally regarded as a booster.

Force exerted by the piston spring of the actuation system during clutch engagement is the same as force on the system during clutch disengagement. This is the combined force of the piston spring and the clutch spring.

This disengagement force (F_d) is equal and opposite to the engagement force (F_e) and is equal to 0.015×10^3 N.

That is

$$F_d = F_e = W_{1\max} = W_{2\max} = 0.015 \times 10^3 \text{ N} \tag{7}$$

Pressure exerted on the outlet or exhaust valve (P_e) is defined as

$$P_e = \frac{\text{Force}}{\text{Area}} \tag{8}$$

$$P_e = \frac{0.015 \times 10^3 \text{ N}}{\frac{\Pi De^2}{4}} = \frac{0.015 \times 10^3 \text{ N} \times 4 \times 10000}{\Pi \times 1.3 \times 1.3} = \frac{6 \times 10^6}{5.31}$$

$$= 1.1299 \times 10^6 \text{ N/m}^2 = 11.299 \times 10^5 \text{ N/m}^2 = 11.299 \text{ bars}$$

The exhaust pressure is thus high enough to ensure that the piston returns to status quo.

The derived data is presented in Table 3.

Table 3. Derived parameters.

S/N	Parameters	Readings
4	Piston coil solid length (L_s)	1.88 cm
6	Piston coil maximum deflection $P(\Delta_L)$	15.12 cm
7	Supply Pressure (P_s)	9.61 Bars
9	Oil pressure	1.61 bars
12	Exhaust Pressure (P_e)	11.299 bars

4. Conclusions

The result obtained from this work is as explicit as expressed in Table 1 and Table 3 above. Table 1 presented the actual measurements while Table 3 presents the derived data. They included the Piston coil diameter (D) which was 4.01 cm while the Piston wire diameter or thickness (d) was 0.235 cm. The Piston coil free length (L_f) was 17 cm while the number of Piston coils (N) was 7. The derived parameters were Piston coil solid length (L_s) which gave 1.88 cm. The Piston coil maximum deflection $P(\Delta_L)$ yielded 15.12 cm. The pressure at the inlet/supply valve or port gave 9.61 bars while 11.299 bars were the derived pressure exerted on the outlet or exhaust valve (P_e).

The study only demonstrates rather an uncommon means of addressing a critical need in the actuation phenomenon. The outcome can safely be adopted in addressing design questions in actuation systems particularly in heavy-duty drive systems in automotive and industrial machine applications.

Acknowledgements

Permit me to acknowledge the critical contributions of the Management and Staff of Tetralog Nigeria Limited Enugu, for permitting me to use their factory and providing the test truck, Mercedes Benz Actros Truck model MP 2 for the field research. In a special way, my thanks go to the Managing Director, Engr. Onuora Nnabugo and His team of Staff including the Service Manager, Engr. E. Obielika, the Assistance Service Manager Engr. D. Gitau and the Training Manager – Mr. C. Nwinyi. The field staff; made up of Mr. J. Nwachukwu, Mr. M. Ogbonna and several others deserve my acknowledgement for their humility and untiring commitment to this project. My regards also go to Engr. G. U. Ezeatu of ANAMMCO Enugu and Engr. Amasha Olakunle of Innoson Industries Limited, Enugu for their contributions. Engr. Dr. J. Osibe, Engr. O. Nwoke, and Mr. R. Iduma, all staff of Akanu Ibiam Federal Polytechnic Unwana are appreciated for allowing me space to use their Strength of Materials Laboratory for this empirical study. My colleagues; Engr. Dr. I.U. Mbabuike, Engr. A.E. Abioye, Mr. J. E. Arua, and Mr. N. Onwuka deserve my thanks for their criticisms and support too.

Conflicts of Interest

The authors declare no conflicts of interest regarding the publication of this paper.

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