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Technical Briefs

Experimental Study of Friction in a Pneumatic Actuator at Constant Velocity

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This paper describes an experimental method of determining sliding friction forces in a pneumatic actuator. Several empirical and semi-empirical friction models are evaluated using measured friction force data. A repeatability study is also performed to qualitatively assess friction randomness and a change in friction regimes.

Introduction

Dynamic friction phenomenon in pneumatic actuation systems is complicated due to the continual transformation between several lubrication regimes, inherent pneumatic system nonlinearities, piston rod binding and misalignments, and manufacturing irregularities, etc. Classical friction theories cannot predict such dynamic friction forces accurately. Consequently, several researchers have suggested empirical or semi-empirical relationships (Gallenstein, 1975; Maalej, 1986;

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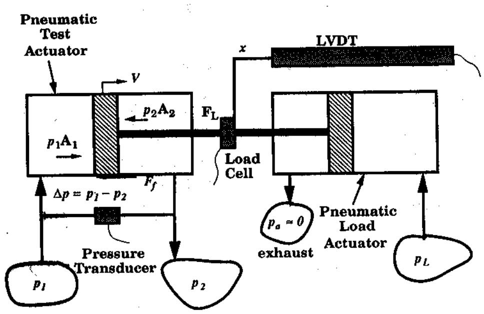

Fig. 1 Experimental setup

Swartzmiller, 1987; Belfort et al., 1990). Nonetheless, experimental methods must be adopted to develop a suitable friction force model. Accordingly, in this paper a technique is outlined for measuring dynamic friction force F_f at a constant actuator velocity V given time-invariant pressure differential Δp across the piston, see Fig. 1 for the schematic. See Singh and Schroeder (1992) for a complete list of equipment.

Design of the Experiment

The pneumatic test cylinder was an off the shelf actuator equipped with a seal-less solid bronze rod bushing. Several aluminum pistons were fabricated and used throughout in the test actuator and light machine oil was used as the lubricant between the piston and cylinder wall. The load actuator was equipped with a Teflon piston so that smaller velocities could be obtained. The friction force magnitude F_f includes both piston-cylinder and rod-seal friction forces while the load force F_L is comprised of all forces acting on the load actuator side of the force transducer. Now define $F_e = \Delta p_e A_p$ as the effective force acting on the piston, where the effective pressure differential is given by $\Delta p_e = p_1(1-\gamma) + \gamma \Delta p$. Here γ is the piston area ratio A_2/A_1 and $\Delta p = p_1 - p_2$. Since inertial forces are negligible, nondimensional friction force is estimated as $\bar{F}_f = F_f/F_e = (1 - (F_L/F_e))$.

Measurements at the inlet port of the test actuator verified that a constant Δp was indeed maintained throughout the forward stroke. Changes in piston velocity V were provided by varying the load pressure p_L, from a source independent of the test actuator sides. Maximum friction force F_f error due to inertial effects was estimated to be less than 8.0×10^{-7} N. Other error analysis results yielded the following: largest deviation in F_L measurement $= \pm 3.60$ N, and the largest possible error in $V = \pm 0.035$ m/s. Measurements were limited however to steady-state conditions for piston velocities V, from 0.05 to 0.63 m/s. Several limitations of the experimental system restricted this range. First, the "stick-slip" behavior at low V was very evident. In addition to the inertial forces associated with piston acceleration during slip, it created difficulty in determining the slope of the velocity curve at the measurement point. Therefore measurements began at a slightly higher V. Second, for large velocities, the coupling force F_L and pressure differential Δp signals were very small. This introduced the possibility of creating large errors during measurement. Third, the amplifiers used in this study caused some signal drift and accuracy was limited by the signal to noise ratio and the digital resolution of the oscilloscope.

Results

In this experiment, nonlinear characteristics of dynamic friction were found to exist for flat-sided pistons since most measurements deviated significantly from the ideal linear viscous damping $F_f = cV$. While the initial viscous damping coefficient c was relatively consistent for different trials, the slope of the

Table 1 Empirical dynamic friction models examined by this study

Model number	Dynamic friction model	Coefficient of determination
A	$F_f = F_o + cV$	0.90
B	$F_f = F_o e^{-\xi V} + cV + \beta \Delta p$	0.91
C	$F_f = F_o + cV + \beta \Delta p$	0.93
D	$F_f = F_o + (1 + cV^n)[\beta \Delta p + \beta_1 p_1]$	0.94
E	$F_f = F_o + cV^n + \beta \Delta p^m$	0.91
F	$F_f = F_o e^{-\xi V} + cV^n + \beta \Delta p^m$	0.95
G	$F_f = F_o e^{-\xi V} + cV + \beta \Delta p + \beta_2 p_2$	0.93

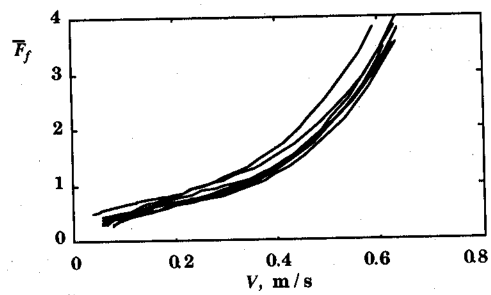

Fig. 2 Measured friction curves for one piston to illustrate repeatability

$F_f(V)$ curve at higher V was usually smaller and was found to be nearly zero in at least one example. Most of the friction curves revealed a sudden regime transition through the mid-velocity range. From thirty-six measurement sets gathered for several different piston geometries and supply pressures, nine representatives sets were chosen for the evaluation of the seven dynamic friction force models as listed in Table 1. To compare the curve-fitting abilities of each model, least-squares regression calculations were performed and the "goodness-of-fit" was determined by finding the corresponding coefficient of determination. The coefficients listed in Table 1 were calculated by averaging the values found from each of the nine data sets. Note that since F_f is clearly a function of both V and Δp, the slope of the $F_f(V)$ curve is not the "classical" viscous damping coefficient (except for Model A). Likewise, F_o represents an asymptotic or bias value, not the "breakaway" or "stiction" friction force.

Model F was found to yield the best fit for six of the nine data sets. The models introducing an exponential term ξ for describing the elastic effect at very low V cannot be evaluated completely here since such measurements at low velocities could not be made. In general, empirically modeling the V and Δp terms with exponents n and m, respectively, resulted in smaller modeling errors, but it increased the computational requirements substantially. Note that Model C (with typical value ranges given by $F_o = -27$ to 31 N, $c = -27$ to 105 N/(m/s), and $\beta = [-0.6$ to $1.9] \times 10^{-04}$ N/Pa) is a special case of Model B and Model E is a special case of Model F when the "inertial and viscous lag" coefficient $\xi = 1$. Also, Model B is a special case of Model F and Model C is a special case of Model E provided $n = m = 1$. It is also interesting to note that Model D was able to fit the experimental data sufficiently well even though it was primarily introduced for conventional pistons with rubber lip seals (Belfort et al., 1990). A linear model of two independent variables represented by Model C was able to fit the data quite well for $F_f(V)$ curves displaying a transformation in the middle of the velocity range. Model G introduces a p_2 term acting across the piston-cylinder rod bearing. The empirical Model E proved to be inferior to other models except for the simplest, but most commonly used linear Model A.

Figure 2 illustrates eight normalized measurement sets gathered for a single piston-cylinder arrangement in order to demonstrate experimental repeatability. The original $F_f(V)$ curves displayed similar transformations, but actual F_f magnitudes varied by as much as 6.7 to 8.9 N for a given velocity. Accordingly, a better relationship is revealed in terms of normalized friction force $\bar{F}_f$. To obtain a useful parameter of friction behavior, a normalized linear damping coefficient $\bar{c}$ is also defined based on eight measured data sets and approximated for two velocity ranges as follows: $\bar{c}=0.0157$ $(\mathrm{m/s})^{-1}$ for $V=0.05$–0.37 m/s, and $\bar{c}=0.0542$ $(\mathrm{m/s})^{-1}$ for $V=0.37$–0.63 m/s. Typically, $\bar{c}$ values were found to be consistent with values from other $F_f(V)$ curves.

Conclusion

In order to establish an applicable dynamic friction model for pneumatic actuators, this study pursued a macroscopic experimental approach. Several empirical models were evaluated, followed by a repeatability study which revealed the random behavior of friction forces. The highly nonlinear empirical Model F was found to have the least error, though simpler models still faired comparatively well. It might be important to note that approximately 75 percent of the thirty-six sets had similar $F_f(V)$ shapes due to regime transformations, though it is uncertain as to why such abrupt changes occur. Further research is needed to address this and other unresolved issues.

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