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Engineering Notes

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Development of a Piezohydraulic Active Pitch Link for a Swashplateless Helicopter Rotor

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Introduction

RECENTLY, there has been considerable interest in achieving swashplateless primary flight control in helicopters. The majority of these studies have investigated on-blade control surfaces such as trailing-edge flaps [1–6] or active twist of the rotor blade [7] to effect a change in the airloads in the rotating frame. The major challenge of these approaches is the design of actuators of sufficiently high power density that are able to operate under large centrifugal loads. To achieve a high-power-density actuation mechanism, most of the on-blade actuation concepts are based on active materials such as piezoelectrics. However, amplifying the limited stroke of active materials to the required levels presents additional practical difficulties [8].

An alternate approach is to incorporate active pitch links (APL) in the rotating frame that change the blade pitch by varying their length. This approach has several advantages. First, the rotor blades are unchanged. Second, as the pitch links are located close to the rotor shaft, they experience a significantly lower centrifugal force than on-blade actuators and do not face stringent volumetric constraints. Third, as the rotor blade does not employ a discrete control surface, the APLs do not create an additional profile drag penalty. Active pitch link concepts have been investigated for individual blade control applications on rotorcraft, both on model-scale [9] and on full-scale rotors. These concepts involve hydraulic actuators mounted on the main rotor hub and consequently require a hydraulic slip ring. As a result, these systems are complicated, difficult to maintain, and result in a large weight penalty. For example, on a system recently tested on the CH-53G helicopter, the pitch link actuators weighed approximately 10 kg, and the overall hydraulic system including the pump, tubing, and slip ring weighed 600 kg [10].

The present study proposes an active pitch link based on a piezohydraulic actuator. By combining the hydraulic pump and actuator into a single unit, this approach will result in a localized self-contained hydraulic actuator that can be mounted in the rotating frame without the need for a hydraulic slip ring. Electrical power will be delivered to the actuators through an electrical slip ring, which already exists in most rotors to service the on-blade deicing units. This will result in a much simpler and lighter actuation system. The implementation of the APL on a rotor blade is shown schematically in Fig. 1. The system consists of a variable-length pitch link in the rotating frame connected to the torque tube of a bearingless rotor blade. The output shaft of a piezohydraulic actuator acts as the variable-length link of the APL.

Actuator Construction

Several researchers have designed and constructed prototype piezohydraulic actuators to establish the proof of concept and to explore the operating principle [11–14]. However, significant challenges remain for practical implementation in an APL system. A key element is to increase the power density of the piezohydraulic actuator, especially by operating at high pumping frequencies. Additionally, the APL must be designed to operate in the rotor environment under centrifugal loads. Although the loads are much lower compared with on-blade actuators, the main rotor hub presents a severe operational environment. As a result, the APL needs to be designed in a way that prevents seizing of the pump under centrifugal loading. The system must also incorporate fail-safe operation by behaving as a passive element in the event of a power loss or actuator failure. A proof-of-concept piezohydraulic actuator specifically designed for compactness and high power density has been developed by the authors and has undergone extensive bench-top testing [15–17]. The geometry of this device is ideally suited for implementation as an APL and will be used in the present study.

A schematic of this device is shown in Fig. 2. The device consists of three main assemblies: a piezoelectric pump, a manifold incorporating an accumulator and directional control valves, and a conventional off-the-shelf hydraulic cylinder. The piezoelectric pump is a hydraulic pump driven by piezoelectric stacks that change their length in response to an applied voltage. The piezoelectric stacks displace a piston, which forces fluid out of the pumping chamber. The oscillatory motion of the piezostacks is rectified into a unidirectional fluid flow by mechanical check valves. This results in a pump with no moving parts, thus significantly simplifying the construction of the actuator and increasing its reliability. The piezoelectric pump supplies pressurized hydraulic fluid to the rest of the circuit at the required flow rate and pressure. Passive reed valves rectify the fluid flow, and the direction of motion of the output shaft is controlled by directional control solenoid valves [18] or magneto-rheological-fluid-based valves [19]. The hydraulic circuit has an accumulator on the low-pressure side, which is used to apply a bias pressure. The high-frequency, low-amplitude displacement of the piezostacks is effectively traded off into a low-frequency, large-amplitude output displacement. As the hydraulic fluid is entirely contained inside the actuator, there is no need for a hydraulic slip

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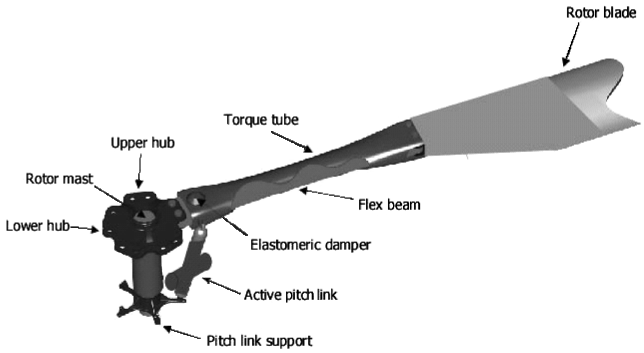


Fig. 1 Conceptual drawing of an active pitch link.

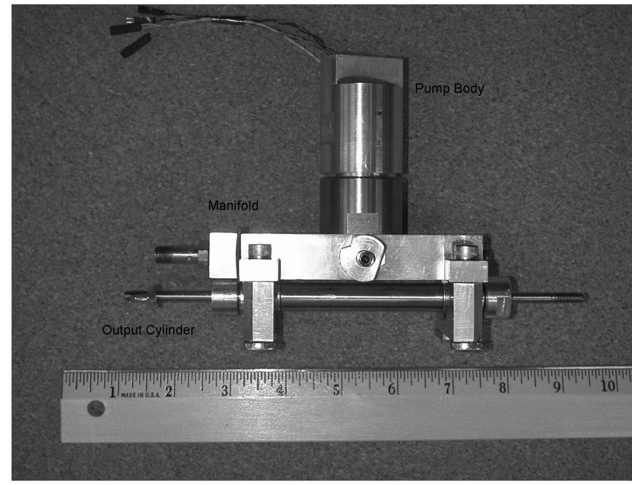
ring. Note that the piezoelectric stacks can be replaced by any type of high-bandwidth active material.

To investigate the effect of centrifugal loads on the performance of the APL, a scaled-down prototype device was tested under rotation in the 10-ft-diam vacuum whirl-test chamber at the University of Maryland. Figure 3a shows a picture of the APL device, and Fig. 3b shows the device on a spin bar with a counterweight at the other end. The radial offset of the APL was approximately 40 cm, and the actuator was tested at rotational speeds of up to 300 rpm, which corresponds to a centrifugal loading of $40g$. Thus, the centrifugal loading experienced by the actuator was nominally higher than the full-scale centrifugal load experienced by a UH-60 pitch link.

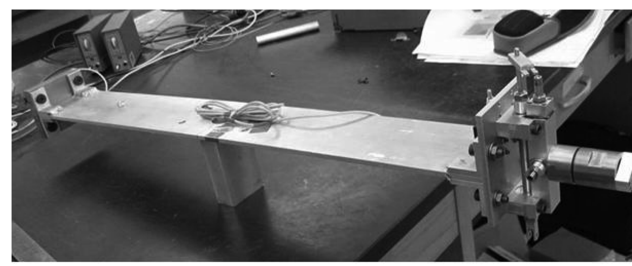
Figure 3a shows the actuator with the manifold and output cylinder. The pump was driven by two Physik Instrumente 804.10 piezostacks and has overall dimensions of a 12×12 mm cross section and 18 mm length with a total weight of 300 g. A low-viscosity hydraulic fluid was used as the working fluid. Under unidirectional operation, a blocked force of 18 lb and a no-load velocity of 5.5 in./s were measured. The salient features of this actuator are listed in Table 1. The hydraulic circuit was pressurized to a bias pressure of 1.4 MPa (200 psi). In the present design, the centrifugal force adds to the preloading of the piezoelectric stack. For simplicity, the actuator was tested without directional control valves under only unidirectional operation.

Experimental Setup

The performance of the actuator was characterized by the velocity of the output cylinder shaft for different values of pumping frequency. An linear variable differential transformer was attached to the output cylinder to measure the velocity of the output cylinder shaft. A spin bar was constructed such that the piezohydraulic actuator was at a distance of 40 cm from the center of rotation. After assembling the actuator, it was attached to the end of the spin bar and balanced with a counterweight (Fig. 3b). The spin bar and the attached actuator system was then installed inside the whirl chamber



a) Picture of a piezohydraulic actuator



b) Bench-top testing of the device on a spin bar

Fig. 3 Piezohydraulic actuator.

of University of Maryland and spun at various angular velocities (Fig. 4). An electrical slip ring transmitted both electrical signals and power to the rotating frame. The data were recorded using a dSPACE data acquisition system.

Two types of tests were performed:

1) In no-load tests, the spin bar was spun at 200 and 300 rpm and the output cylinder shaft velocity was measured over a range of pumping frequencies, which are the frequencies at which the piezostacks are actuated. The maximum output velocity of the actuator was obtained during no-load testing as energy was lost only in overcoming the internal losses in the actuator.

2) In load tests, the actuator was made to work against external load springs of known stiffness (12 and 18 lb/in.). The values of spring stiffness were chosen to provide a force equivalent to the blocking force of the actuator at the full stroke of the output cylinder. The full stroke of the output cylinder was 2 in. and the blocked force of the actuator was calculated to be around 35 lb. Thus, the spring with a spring constant of 18 lb/in. provided 36 lb of force at a stroke of 2 in. The performance of the actuator was measured as the velocity of

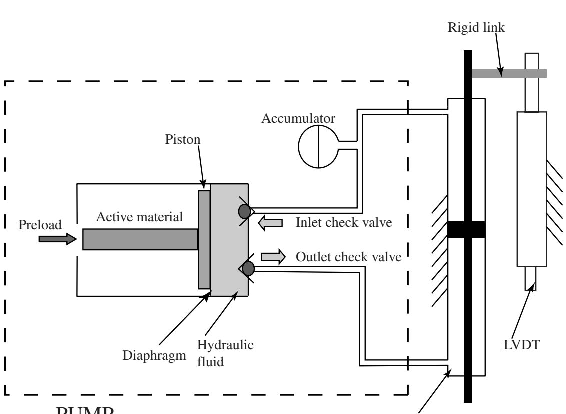


Fig. 2 Schematic diagram of the components of the hybrid hydraulic actuator (LVDT denotes the linear variable differential transformer).

Table 1 Pump properties

Pump body assembly	
Pump body diameter, mm, in.	35.6, 1.4 o.d., 25.4, 1 i.d.
Pump body length, mm, in.	50.8, 2
Piston diaphragm thickness, mm, in.	0.05, 0.002
Pumping chamber diameter, mm, in.	25.4, 1
Pumping chamber height, mm, in.	0.76, 0.030
Valve assembly	
Valve plate thickness, mm, in.	5.1, 0.2
Reed valve thickness, mm, in.	0.051, 0.002
Hydraulic circuit	
Accumulator gas volume, mm^3, $in.^3$	1638.7, 0.1
Output cylinder bore, mm, in.	11.1, 7/16
Output shaft diameter, mm, in.	1.58, 3/16
Output cylinder stroke, mm, in.	50.8, 2

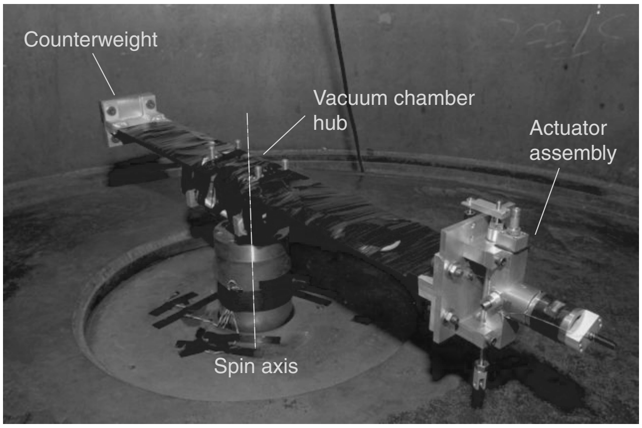


Fig. 4 Photograph showing the spin bar with counterweight and actuator assembly installed inside the whirl chamber.

the cylinder shaft at different pumping frequencies based on the actuator resonance obtained from no-load tests.

Results and Discussion

The primary effect of centrifugal forces is increased friction in the output cylinder. Bearings support the cylinder shaft and shaft piston inside the output cylinder. Hence, the performance of the actuator is linked to the performance of the bearings. The performance of the actuator was measured for two kinds of bearings: sliding bearings and roller bearings.

Unidirectional No-Load Tests with Sliding Bearings

In the off-the-shelf hydraulic cylinder, the cylinder shaft was supported on two sliding bearings. The test results using this cylinder are shown in Fig. 5. As we can see from the figure, there was a sharp drop in performance with increasing angular velocity. At the resonant pumping frequency of 800 Hz, there is a reduction of more than 50% in the output shaft velocity. Moreover, there were numerous fluid leaks around the output cylinder.

Because of the pump piston, the centrifugal force results in an increase in preload on the piezoelectric stacks. The net force that the piston (of mass 30 g) exerts on the piezostacks was calculated as 12 N. Because this additional force was only 0.24% of the block force of the stack (which is 5000 N), it was concluded that this effect was not responsible for the drastic drop in performance. The only possible reason for the drastic change in performance was determined to be the increase in friction in the output cylinder due to centrifugal forces. To alleviate this problem, a new output cylinder was designed with roller bearings instead of sliding bearings.

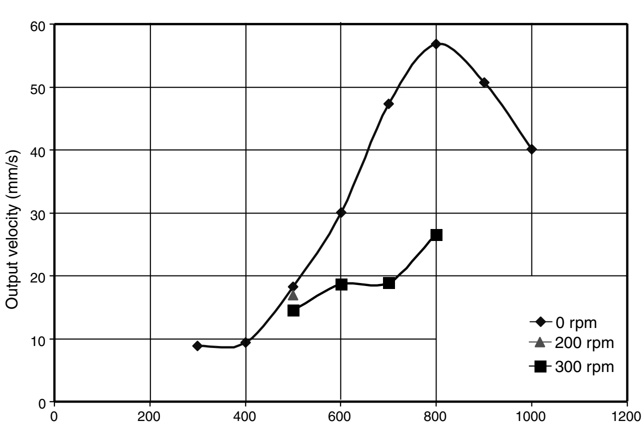


Fig. 5 Results of preliminary whirl tests.

Unidirectional No-Load Tests with Roller Bearings

These tests were performed using an actuator that was powered by three piezostacks. A custom-designed output cylinder with roller bearings was fabricated and tested in the experimental setup. The new output cylinder had an outer diameter of $\frac{3}{4}$ in. (19.05 mm), an inner diameter of $\frac{1}{4}$ in. (6.35 mm), and a stroke of 2 in. (50.8 mm). Because the output cylinder has a different area of cross section compared with the earlier output cylinder, there is no direct comparison between the results. The criterion of comparison will be the percentage drop in performance with angular velocity. The results of the whirl tests are shown in Fig. 6. As we can see from the figure, there is no appreciable drop in performance with increasing angular velocity. All the three curves are within the error bars of the baseline curve. Thus, it is ascertained that the reason for the drop in performance in the preliminary tests was the increased friction in the output cylinder shaft.

Load Tests

These tests were performed to ascertain the performance of the hybrid hydraulic actuator under load in a rotating environment. The performance of the actuator under spring load was analyzed. Preloaded springs were attached to the output cylinder shaft such that as the cylinder shaft moves, the springs are extended, thus forcing the actuator to work against the spring load. Two springs with different spring constants were used in this test. Spring 1 had a spring constant of 2000 N/m (12 lb/in.) and spring 2 had a spring constant of 3150 N/m (18 lb/in.). Figure 7 shows the actuator with one of the

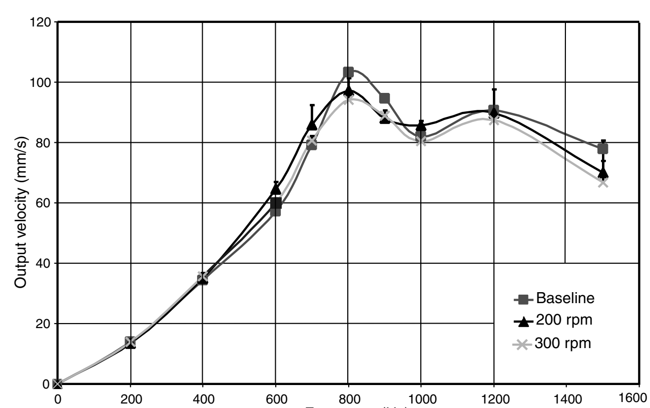


Fig. 6 Results of whirl testing with improved output cylinder design.

Fig. 7 Photograph of the actuator working against a stiffness of 18 lb/in..

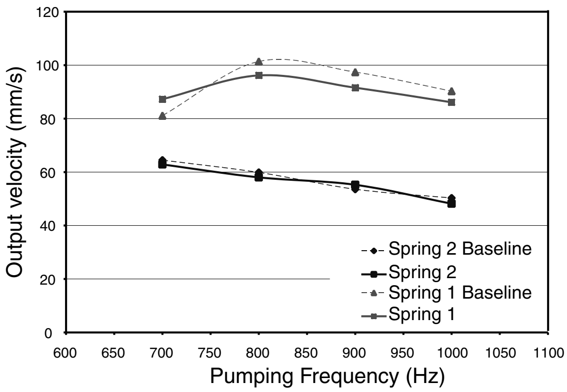


Fig. 8 Results of whirl testing with stiffness load.

attached springs. Baseline data were taken when the spin bar was stationary for both cases of spring constants. The actuator was then spun at 300 rpm (approximately $40g$ centrifugal loading) and data were recorded for four values of pumping frequencies around the actuator resonance (700–1000 Hz pumping frequency). The intention was to observe any deterioration in performance at and around the pumping frequency with the maximum performance (as seen from no-load tests). The results from the whirl testing are shown in Fig. 8. We can see from the figure that there is no deterioration in the actuator load performance in a rotating environment for either of the springs used. Thus, we can conclude that the present actuator configuration is able to maintain baseline performance for a spring-loaded case of 3150 N/m and a centrifugal loading of up to $40g$ ($392\ \mathrm{m/s^2}$).

Conclusions

The feasibility of operating a piezohydraulic actuator as an active pitch link in conditions of high centrifugal loading is evaluated. Because the intended application in this study was a pitch link, a scaled-down prototype actuator was tested at a centrifugal load of $40g$, which is in excess of the full-scale load experienced by a UH-60 pitch link. Preliminary tests showed a deterioration in performance with increasing centrifugal force due to increased friction in the output cylinder shaft. A new output cylinder was designed that incorporated roller bearings to support the centrifugal force on the cylinder shaft. Tests of this improved design showed no deterioration in performance of the actuator up to the full-scale loading condition.

The performance of the actuator was also evaluated under an external spring load. It was found that the performance of the actuator did not deteriorate from the baseline case. Thus, the existing design of piezohydraulic actuator was able to maintain its performance for loaded and no-load cases under a full-scale centrifugal loading environment. Future work will involve the investigation of scaling effects on the piezohydraulic actuator to enable operation under full-scale pitch link loads.

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