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POWER CONTROL UNITS WITH SECONDARY CONTROLLED HYDRAULIC MOTORS – A NEW CONCEPT FOR APPLICATION IN AIRCRAFT HIGH LIFT SYSTEMS

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ABSTRACT

Today's high lift systems of civil transport aircraft are driven by Power Control Units using valve controlled constant displacement hydraulic motors. This concept leads to complex valve blocks, attended by high power losses to realise discrete speed control, positioning and pressure maintaining functionality. The concept of secondary controlled hydraulic motors with variable displacement offers reduction in flow consumption without pressure losses and decreases the complexity of the valve block design. Instead of controlling flow of the hydraulic motor with valves, torque is adjusted to the load by varying displacement. An electronic control circuit allows flexible digital control concepts e.g. load independent speed control, pressure maintaining functionality, smooth start-up sequences and continuous positioning of the mechanical transmission system.
This paper introduces the concept of today's Power Control Units, the principle and mathematical model of secondary controlled hydraulic motors and the cascade control loop structure. A new hydraulic concept for Power Control Units using secondary controlled hydraulic motors is presented. Theoretical, simulated and experimental results show typical operation sequences under load and a comparison of power requirement to conventional systems.

KEYWORDS

Secondary controlled hydraulic motor, variable displacement hydraulic motor, aircraft high lift system, secondary flight controls, power drive units, Power Control Unit (PCU)

INTRODUCTION

The power requirements of future large civil transport aircraft open an attractive field for the application of secondary controlled hydraulic or so called variable displacement hydraulic motors (VDHM). Especially during landing approach, the operation of power drive units of high lift systems, so called Power Control Units (PCU), heavily loads the hydraulic power supply (Ivantysynova *et al*, 1995).
Figure 1 shows a typical load profile for a civil transport aircraft hydraulic system. The consumption during the approach phase is one decisive design case for today's aircraft hydraulic systems sizing. In this flight phase, large hydraulic consumers (flaps/slats, landing gear) have to be operated while the available hydraulic power pump flow reaches its minimum due to idle condition of the engines.

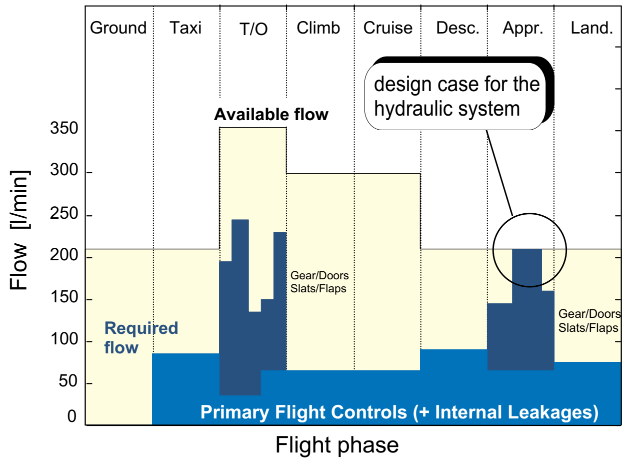

Figure 1. Typical hydraulic load profile

A new concept of PCU design using secondary controlled hydraulic motors with variable displacement reduces the flow demand from the hydraulic system during take-off and landing (Geerling, 1997). Compared to conventional PCU, a flow reduction of about 50% is expected.
This technology is under investigation and development at the Section Aircraft Systems Engineering at the Technical University of Hamburg-Harburg in cooperation with Liebherr Aerospace Lindenberg. For that, a test set-up and comprehensive simulation models were established to develop the new concept.

I POWER CONTROL UNIT

Today's high lift systems of civil transport aircraft are driven by PCU using valve controlled constant displacement hydraulic motors (CDHM). Figure 2 shows a typical high lift transmission system with a conventional PCU of the leading edge (slats). The same principle is not shown but also applied for the trailing edge (flaps). For reliability aspects the PCU has two independent hydraulic motor/valve group assemblies. A speed summing differential gear (DG) connects both hydraulic motors to the transmission shaft. In case of a single hydraulic system failure the slats resp. flaps are operated with half speed. The feedback position pick-up unit (FPPU) indicates the position of the transmission system. If the desired flap position is reached, the whole transmission system is set by applying pressure-off brakes (POB).

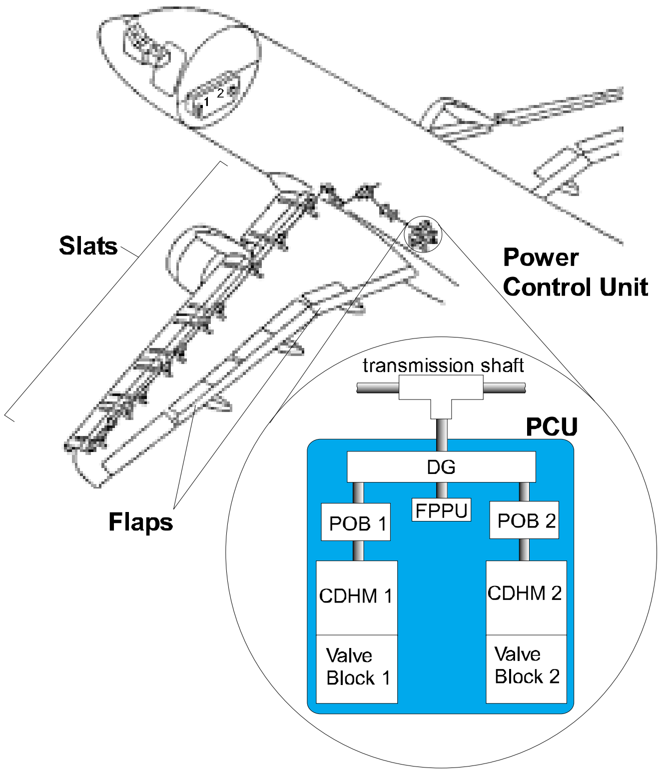

Figure 2. Conventional high lift system with CDHM-driven PCU

Figure 3 illustrates a detailed example of the hydraulic scheme of one CDHM with valve block as it is applied in the Airbus A340. Different hydro-mechanic control functions are realised. Direction and two discrete rates of speed are controlled by the main control valve and a pilot flow limiting restrictor. A pressure maintaining function, using a pilot pressure maintaining valve is included to give priority to primary flight controls, reducing flow consumption of the motor if the system pressure drops. The flaps are positioned by depressurising the POB with the brake solenoid valve (Brake SV). The functions are thus reached by switching several discrete solenoid valves (SV).
The implemented control functions require a complex valve block design. This kind of speed control, by varying the hydraulic resistance, leads to pressure losses up to 80 %.

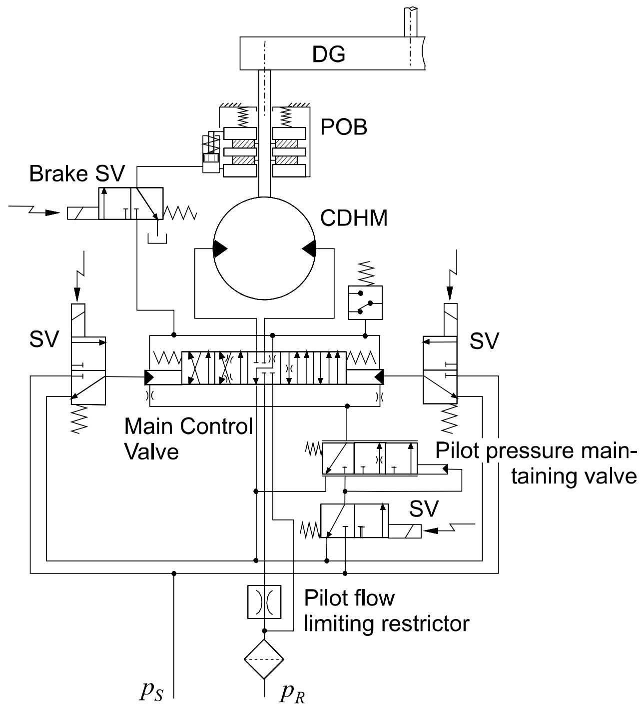

Figure 3. Hydraulic concept of a valve controlled CDHM

II SECONDARY CONTROLLED HYDRAULIC MOTORS

The principle of secondary controlled hydraulic units or so called variable displacement hydraulic motors (VDHM) allows conversion of hydraulic to mechanical power without pressure losses. It has been successfully applied in a variety of industrial fields since the early eighties. The use in aircraft's hydraulic systems requires reliability and safety under extreme environmental conditions and life time demands.

2.1 Design and Function

Figure 4 shows the principle and a cross section of an axial piston motor. The motor torque is regulated by the angle of the swash plate, changing the motor displacement. It is positioned by a swash plate actuator (SPA) which is controlled by an electro-hydraulic servovalve (EHSV).

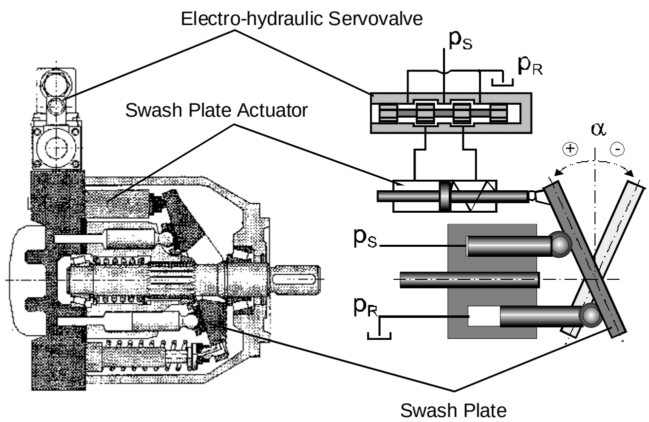

Figure 4. Design of an axial piston motor (Mannesmann Rexroth Brueninghaus Hydromatik, type A10VSO)

The design described allows a very flexible application of VDHMs in aircraft hydraulic architecture (Biedermann *et al*, 1998). Depending on the swash plate angle and the load torque at the output shaft the unit works either as a pump or as a motor in 4-quadrant mode. This kind of hydraulic motor allows control of torque, power, speed and position at the output shaft.

2.2 Model of the Hydraulic Motor

This section presents a non-linear mathematical model of the hydraulic unit. Figure 5 shows a scheme of a VDHM with constant pressure supply.

The secondary controlled hydraulic motor can be characterised by two equations, namely for flow and motor torque depending on the variable displacement V_M. The motor torque M_M at constant differential pressure supply is

$$M_M = V_M \cdot \frac{(p_S - p_R)}{2\pi} \tag{1}$$

with the displacement V_M being proportional to the swash plate actuator stroke x_P. This leads to

$$M_M = x_P \cdot \frac{V_{M,max}}{x_{P,max}} \frac{(p_S - p_R)}{2\pi} . \tag{2}$$

The motor flow Q_M for a constant speed ω is given by

$$Q_M = V_M \cdot 2\pi \cdot \omega = x_P \cdot \frac{V_{M,max}}{x_{P,max}} \cdot 2\pi \cdot \omega . \tag{3}$$

The hydraulic input power of a VDHM $P_{hyd,in}$ at constant differential pressure supply

$$P_{hyd,in} = (p_S - p_R) \cdot Q_M \tag{4}$$

is only reduced by the hydro-mechanical efficiency η_{hm} and volumetric efficiency η_{vol}. Hence, the mechanical power $P_{mech,out}$ at the output shaft is calculated by

$$P_{mech,out} = M_L \cdot \omega = P_{hyd,in} \cdot \eta_{hm} \cdot \eta_{vol} = P_{hyd,in} \cdot \eta_t \tag{5}$$

with the load torque M_L at the motor output shaft resp. at the PCU output shaft, considering aerodynamic loads and mechanical losses of the transmission system.

The power equation (5) shows, that power loss of a VDHM only depends from the volumetric and hydro-mechanical efficiencies. VDHM are used to work with an overall, total efficiency η_t up to 90% at the operation point.

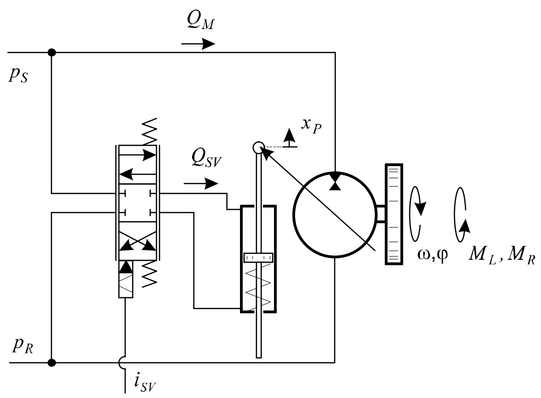

Figure 5. Scheme of a VDHM

A non-linear mathematical model of the hydraulic motor is used to simulate dynamic behaviour and for controller design. The mechanical system of the VDHM is described by the equation of momentum

$$J \cdot \dot{\omega} = M_M - M_F - M_L . \tag{6}$$

The friction term M_F considers Coulomb and viscous friction as well as the stiction moment:

$$M_F = M_{Coul}\operatorname{sign}(\omega) + d \cdot \omega + M_{Stic}\operatorname{sign}(\omega) \cdot \mathrm{e}^{-\frac{|\omega|}{\omega_0}} . \tag{7}$$

Equations (2), (6) and (7) lead to

$$J \cdot \dot{\omega} + d \cdot \omega = x_P \cdot \frac{V_{M,max}}{x_{P,max}} \frac{(p_S - p_R)}{2\pi} - M_L - \tag{8}$$

$$- M_{Coul}\operatorname{sign}(\omega) - M_{Stic}\operatorname{sign}(\omega) \cdot \mathrm{e}^{-\frac{|\omega|}{\omega_0}}$$

a non-linear, time-invariant ordinary first order differential equation with the actuating input x_P. The system dynamic depends on the difference between motor torque M_M (x_P) and load torque M_L. Simplified the swash plate actuator is represented by an integral behaviour

$$\dot{x}_P = \frac{Q_{SV}}{A_P} \tag{9}$$

and the flow through the EHSV by a proportional term

$$Q_{SV} = K_{SV} i_{SV} , \tag{10}$$

with a linear approximation for the servovalve gain K_{SV}.

III NEW CONCEPT FOR AIRCRAFT APPLICATION

The use of VDHM in PCU application leads to new hydraulic interface and controller concepts. Therefore, a test set-up and simulation model was established at the Section Aircraft Systems Engineering of the Technical University of Hamburg-Harburg to examine the new concept in practical operation.

3.1 Hydraulic Concept

Figure 6 shows a new possible configuration for a VDHM-driven PCU in 4-quadrant mode. The hydraulic unit is separated from the pressure supply by an isolation or so called enable valve during non-operational time. In combination with the Brake SV it ensures adequate protection against failure cases which might lead to uncommanded flaps movement or runaway. In pump mode, i.e. for 'aiding load' cases, the VDHM inlet and suction port is connected to the system return pressure line and pump flow is swept via a pressure relief valve to the return pressure line.

3.2 Controller Design

The VDHM concept with continuous control of motor torque allows a flexible and application specific integration of different modes as speed control, start-up and positioning sequences and pressure maintaining function into the closed loop high lift positioning circuit. Compared to the conventional CDHM in Figure 3 the functions of the main control valve, pilot flow limiting restrictor and pilot pressure maintaining valve can be realised by the digital controller using signals of motor shaft speed ω, swash plate position x_P and system pressure p_S.

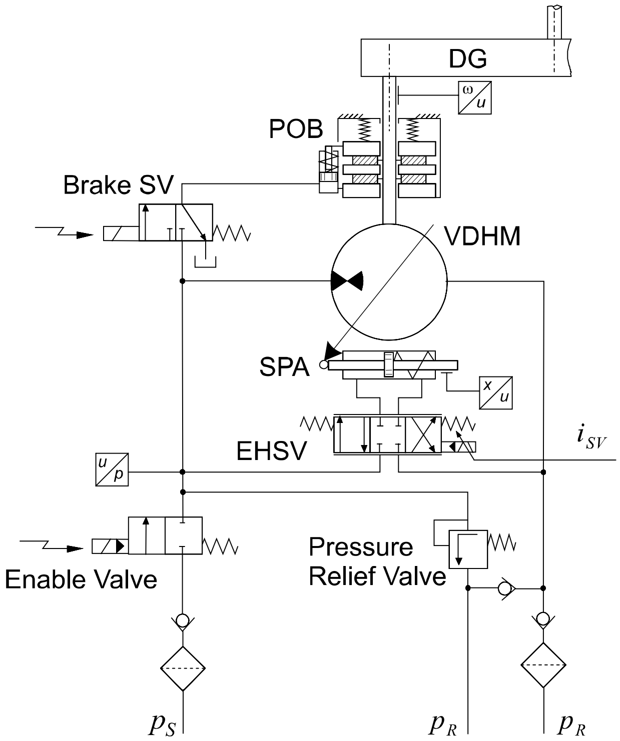

Figure 6. Hydraulic concept of PCU with VDHM in 4-quadrant operation

Figure 7 illustrates a principle controller structure. Swash plate stroke, speed and transmission position are controlled in a cascade control loop design. Each VDHM is associated to one controller. Depending on the difference between transmission position φ_{FPPU} and desired position φ_{in} a speed trajectory ω_{in} is defined. The pressure maintaining function (PMT) affects speed limiting if the system pressure p_S drops under a certain limit.

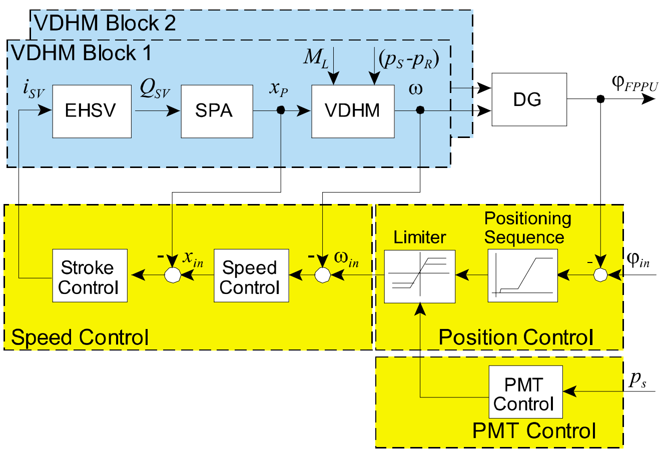

Figure 7. Block diagram of the cascade control loop design

3.3 Experimental Results

The test set-up consists of an Airbus A310 DG driven by two secondary controlled axial piston VDHM as shown in Figure 4. Loads at the output shaft of the DG are simulated by a servovalve controlled constant displacement motor. The cascade control concept was realised as digital controller and executed on a personal computer (Geerling, 1997).

Figure 8 demonstrates a comparison of simulated and measured results for swash plate angle resp. displacement and speed during a full flap extension assuming an Airbus A340 load profile. For simulation the non-linear model mentioned before was used.

The start-up sequence releases the POB with a simultaneous input speed ramp being applied. The shaft speed ω shows load independent behaviour. The displacement indicated by the piston stroke x_P adjusts to the changing load. At max. load conditions just 60% of max. displacement is needed. The difference between required and max. displacement directly shows the power reduction between constant (max.) displacement and variable displacement hydraulic motors. When the desired flap position is approached, a shut down sequence is initiated and the POB is set.

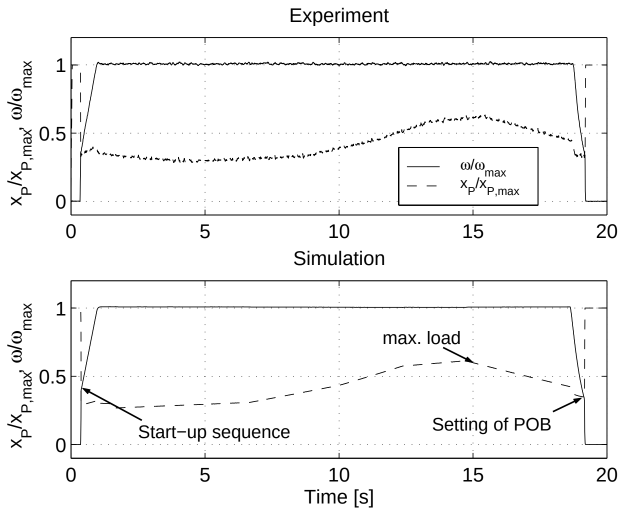

Figure 8. Comparison of experiment and simulation with VDHM (flap extension against load)

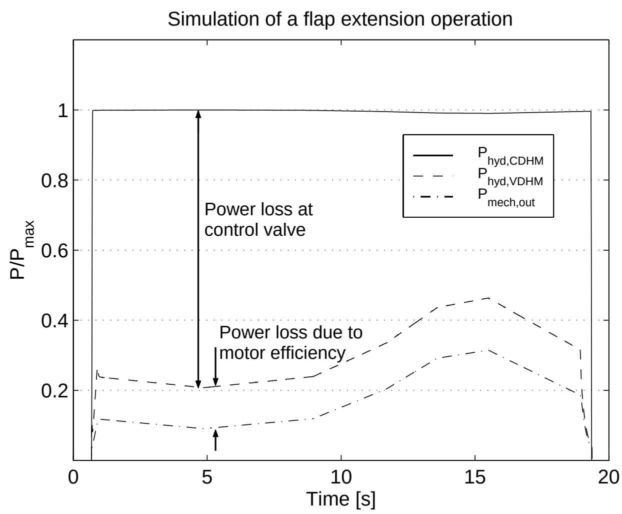

Figure 9. Comparison of power need between CDHM and VDHM

3.4 Hydraulic Power Consumption

Figure 9 shows the theoretical, simulated power consumption of a conventional CDHM-driven and the new VDHM-driven PCU concept. A comparison for a typical flap extension based on Airbus A340 data is made. The given load profile at the PCU output shaft considers aerodynamic loads as well as mechanical losses of the transmission system. This load profile multiplied with the actuated speed leads to the mechanical output power $P_{mech,out}$ at the PCU output shaft.

The power consumption of the VDHM-driven PCU $P_{hyd,VDHM}$ adapts to the changing output power $P_{mech,out}$ and compensates losses due to the total efficiency η_t of the motor. The flow controlled CDHM-driven PCU has a constant power consumption $P_{hyd,CDHM}$ assuming steady speed. The power difference between $P_{hyd,CDHM}$ and $P_{hyd,VDHM}$ is caused by pressure losses in the main control valve and the flow limiting restrictor.

The comparison shows a possible power reduction between 53% and 80% applying VDHM instead of CDHM in the A340 PCU.

CONCLUSION

Introducing variable displacement to power drive units offers a high potential for hydraulic system power reduction.

At the Section Aircraft Systems Engineering of the Technical University of Hamburg-Harburg a first step to investigate this technique has been applied for the PCU of high lift systems. Different hydraulic concepts and controller designs have been investigated. Feasibility, practicability and reliability have been proved by experimental, numerical and analytical results.

This paper has presented theoretical and experimental results on a new concept for application in aircraft high lift systems. The comparison to today's conventional system verifies power reduction between 53% and 80%. Moreover, valve block complexity is decreased. A digital controller allows flexible transfer of hydro-mechanical control functions and offers all kinds of speed, position, torque or power control for future concepts.

The consequent use of hydraulic motors with variable displacement in aircraft's hydraulic system architecture could decrease system power requirements. The principle is transferable to any other consumers with rotary power drive units, e.g. Trimmable Horizontal Stabiliser Actuator.

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NOTATIONS

Symbols

A	Surface area
J	Inertia
K	Gain
M	Torque
P	Power
Q	Flow
V	Displacement
d	Viscous friction number
i	Current
p	Pressure
x	Actuator stroke
ω	Revolving shaft speed
φ	Transmission position
η	Efficiency

Indices and Abbreviations

CDHM	Constant displacement hydraulic motor
Coul	Coulomb
DG	Differential gear
EHSV	Electro-hydraulic servovalve
F	Friction
FPPU	Feedback position pick-up unit
hm	hydro-mechanical
hyd	hydraulic
in	Input
L	Load
M	Hydraulic motor
mech	mechanical
out	Output
P	Piston
PCU	Power Control Unit
PMT	Pressure maintaining
POB	Pressure-off brake
R	Return
S	Supply
SPA	Swash plate actuator
Stic	Stiction
SV	Servovalve, Solenoid valve
t	total
VDHM	Variable displacement hydraulic motor
vol	volumetric

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