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PERFORMANCE OF A POWER TRANSFER UNIT FOR AIRCRAFT APPLICATIONS

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ABSTRACT

This paper considers power transfer between two fluid power circuits such that a failing pressure in one of them is improved by power transfer from the other circuit. The two coupled axial piston machines are effectively driven by the pressure differential across the two machines and particular features are required for the unit to operate successfully. Steady state theory is presented that may be used to select the relative displacement ratios and the range predicted is compared with those typically found in aircraft applications.

KEY WORDS

PTU, pump, motor, direct coupling

NOMENCLATURE

B_v	combined viscous friction coefficient
D_m, D_p	motor and pump displacements
P_1 , P_2	pressure at load a and load b
P_{rv}	pressure relief valve setting
Q_1 ,Q_2	flow into and out of the PTU
Q_a ,Q_b	flows to load a and load b
Q_{p1} ,Q_{p2}	flows from supply pump 1 and pump 2
Q_{po}	ideal flows from both supply pumps
Q_{rv1}, Q_{rv2}	flows through PRV1 and PRV 2
R_{mp}	PTU motor and pump equal resistance
R_{ps}	supply pumps equal resistance
R_t	total resistance
R_v	PRV resistance for both valve
T_{sc}	total static/coulomb friction torque
ε	displacement ratio D_p /D_m
ω	PTU speed

INTRODUCTION

This approach is used to ensure that pressure is maintained in a circuit in the event of an unacceptable or unexplained drop in pressure, and is used in aircraft applications. Another existing healthy circuit is used to supply flow to the faulty circuit in a manner that attempts to restore the faulty pressure as close as possible to its normal healthy value. The approach is shown in figure 1. For the purpose of example, healthy circuit **a** provides make-up flow rate to faulty circuit **b** via the Power Transfer Unit (PTU), the left hand side of the PTU acting as a motor and the right hand side of the PTU acting as a pump. The PTU rotates due to the net pressure differential across the motor unit being greater than the net pressure differential across the pump unit. This creates a torque unbalance beyond the friction value when one circuit pressure changes due to a fault condition. In practice, a variable displacement axial piston machine is used together with a bent axis fixed displacement piston machine. The machine acting as a motor must have a displacement greater than the machine acting as a pump. As the direction of power transfer is changed then the swash-plate stroke of the

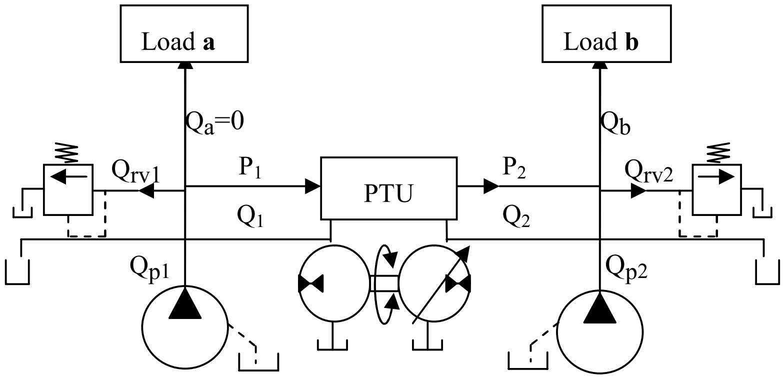

Figure 1 Power transfer using reversible axial piston units

variable displacement machine must also be changed. Power transfer is from left to right due to a pressure drop in supply line 2 relative to supply line 1. Therefore the machine at the left hand side is acting as a motor and the machine at the right hand side is acting as a pump.

There is little to no published information on PTUs and the following analysis is based upon well established steady state pump/motor theory. Some SAE and patent information may be found in [1-7]. It will be deduced for patent information that there are some complex mechanisms developed to create displacement ratio change, and the dynamic behaviour of a PTU is also complex. The steady state theory presented here has resulted from the author's involvement with a particular manufacturer, but precise details cannot be disclosed. The approach is therefore generalized, but results are compared with typical SAE data.

THEORY

Consider therefore the following theory [8-11] which is satisfactory for a first-estimate of performance :

PTU flow rates Motor
$$Q_1 = D_m \omega + \frac{P_1}{R_{mp}} \quad (1)$$

Pump
$$Q_2 = D_p \omega - \frac{P_2}{R_{mp}} \quad (2)$$

PTU torque
$$D_m P_1 - D_p P_2 = B_v \omega + T_{sc} \quad (3)$$

Here the viscous coefficient B_v and the friction torque T_{sc} is the sum for both machines. Considering each identical supply pump :

$$Q_{p1} = Q_{po} - \frac{P_1}{R_{ps}} \quad (4)$$

$$Q_{p2} = Q_{po} - \frac{P_2}{R_{ps}} \quad (5)$$

Assuming that both identical PRVs are in operation on both power supply sides then a linear flow rate/pressure drop characteristic may be used to give :

$$Q_{rv1} = \frac{P_1 - P_{rv}}{R_v} \qquad P_1 > P_{rv} \quad (6)$$

$$Q_{rv2} = \frac{P_2 - P_{rv}}{R_v} \qquad P_2 > P_{rv} \quad (7)$$

For the purpose of identifying unique features of a PTU performance, a suddenly-demanded flow rate Q_b at side 2 will be used while the load flow rate at side 1 is zero. Therefore, flow rate continuity on each side then gives :

$$P_1 = R_t \left(Q_{po} - D_m \omega + \frac{P_{rv}}{R_v}\right) \quad (10)$$

$$P_2 = R_t \left(Q_{po} + D_p \omega + \frac{P_{rv}}{R_v} - Q_b\right) \quad (11)$$

$$\frac{1}{R_t} = \frac{1}{R_{ps}} + \frac{1}{R_{mp}} + \frac{1}{R_v} \quad (12)$$

It is well established that a PTU performs better if the pump displacement is less than the motor displacement, so let :

$$D_p = \varepsilon D_m \qquad \varepsilon < 1 \quad (13)$$

Combining the torque and flow equations and the displacement ratio equation then gives :

$$\overline{\omega}=\frac{(\overline{P}_{rv}+1)(1-\varepsilon)+\varepsilon\overline{Q}_b-\overline{T}_{sc}}{(1+\varepsilon^2+\overline{B}_v)} \quad (14)$$

$$\frac{P_1}{P_{rv}}=\frac{R_t}{R_v}\left[\frac{(1-\overline{\omega})}{\overline{P}_{rv}}+1\right] \quad (15)$$

$$\frac{P_2}{P_{rv}}=\frac{R_t}{R_v}\left[\frac{(1+\varepsilon\overline{\omega}-\overline{Q}_b)}{\overline{P}_{rv}}+1\right] \quad (16)$$

where the non-dimensional term are :

$$\overline{\omega}=\frac{D_m\omega}{Q_{po}} \quad \overline{T}_{sc}=\frac{T_{sc}}{D_m R_t Q_{po}}$$

$$\overline{P}_{rv}=\frac{P_{rv}}{R_v Q_{po}} \quad \overline{Q}_b=\frac{Q_b}{Q_{po}} \quad \overline{B}_v=\frac{B_v}{R_t D_m^2}$$

$$\frac{R_v}{R_t}=\frac{R_v}{R_{ps}}+\frac{R_v}{R_{ps}}+1 \quad (17)$$

where probably $R_v \cong R_t$

The individual pressures are defined with respect to the pressure relief valve cracking pressure P_{rv} so a check can be made on whether a particular design causes any pressure to fall below the pressure relief valve setting. The pressure differential must lie outside the friction dead-band for PTU motion to occur. This is given by :

$$\text{PTU friction dead-band} \quad \overline{P}_1-\varepsilon\overline{P}_2=\pm\frac{\overline{T}_{sc}}{\overline{P}_{rv}} \quad (18)$$

Consider the following, real, data :

PTU machines $R_{mp} = 10^{12}$ Nm^{-2} /m^{3}s^{-1}

Supply pumps $R_{ps} = 10^{12}$ Nm^{-2} /m^{3}s^{-1}

PRVs $R_v = 0.25\text{x}10^{10}$ Nm^{-2} /m^{3}s^{-1}

PRV cracking pressure P_{rv} = 210bar

Supply pumps no-load flow rate Qpo = 24litres/min

Fixed displacement machine displacement

$D_m = 4\text{x}10^{-6}$ m^3/rad

PTU friction torque T_{sc} = 12Nm

PTU viscous friction coefficient B_v = 0.04 Nm/rad s^{-1}

$$\frac{1}{R_t}=\frac{1}{R_{ps}}+\frac{1}{R_{mp}}+\frac{1}{R_v}=402\text{x}10^{-12}$$

$$R_t \cong R_v = 0.25\text{x}10^{10}\,\text{Nm}^{-2}/\text{m}^3\text{s}^{-1}$$

The PRV resistance is clearly dominant. For no-load flows from each power supply the pressure increase across each PRV is given by :

$$R_v\, Q_{po} = (0.25\text{x}10^{10})(0.4\text{x}10^{-3}) = 10\text{bar}$$

The non-dimensional parameters are :

$$\overline{P}_{rv}=21 \quad \overline{B}_v=1 \quad \overline{T}_{sc}=3$$

If the PRV cracking pressure is 210bar, then the operating pressure for no-load is not 220bar but ≈ 219bar when the PTU losses are included.

To fully understand how a PTU operates it is necessary to consider particular conditions of speed and pressure.

The condition for zero speed

From (14) this occurs when the following condition is satisfied :

$$\varepsilon=\frac{\overline{P}_{rv}+1-\overline{T}_{sc}}{\overline{P}_{rv}+1-\overline{Q}_b} \quad (19)$$

The condition for pressures to fall to their PRV setting.

To understand the pressure behaviour, it is noted that since it is probable that $R_t \approx R_v$ then the pressures are given by:

$$\frac{P_1}{P_{rv}}\approx\frac{(1-\overline{\omega})}{\overline{P}_{rv}}+1 \quad \frac{P_2}{P_{rv}}\approx\frac{(1+\varepsilon\overline{\omega}-\overline{Q}_b)}{\overline{P}_{rv}}+1 \quad (20)$$

Therefore the condition for each pressure to fall to the PRV setting is given by :

$$\frac{P_1}{P_{rv}}=1 \;\; \text{when} \;\; \overline{\omega}=1$$

$$\varepsilon^2+\varepsilon(\overline{P}_{rv}+1-\overline{Q}_b)-(\overline{P}_{rv}-\overline{B}_v-\overline{T}_{sc})=0 \quad (21)$$

Therefore the speed is at its maximum

$$\frac{P_2}{P_{rv}}=1 \;\text{when}\; 1+\varepsilon\overline{\omega}=\overline{Q}_b \;\text{and not possible} \quad (22)$$

The example conditions are shown in figure 2 for various load flows and figure 3 for a particular load flow $\overline{Q}_b = 0.5$.

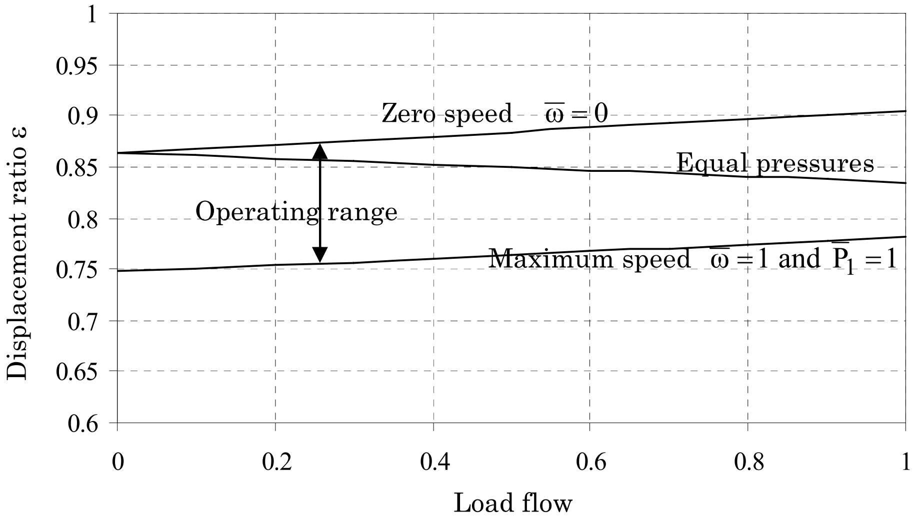

Figure 2 Determination of the PTU displacement ratio

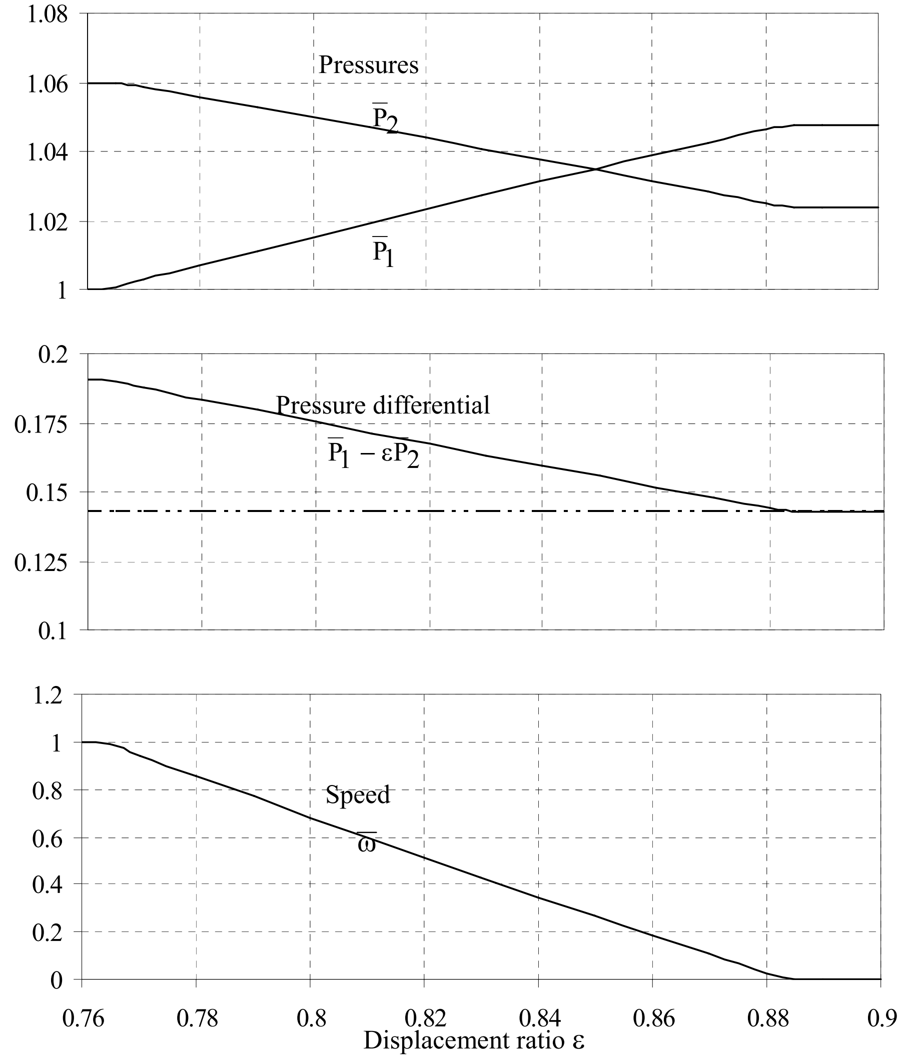

Figure 3 PTU performance for a specific undesirable load flow $\overline{Q}_b = 0.5$

The condition for equal pressures.

It seems reasonable to aim for equal pressures under PTU operation as a good design guide. This condition is achieved when :

$$\overline{P}_1 = \overline{P}_2 \text{ when } \overline{\omega} = \frac{\overline{Q}_b}{(1+\varepsilon)}$$

$$\varepsilon^2(\overline{P}_{rv}+1)+\varepsilon(\overline{T}_{sc}-\overline{Q}_b) + \overline{T}_{sc}+\overline{Q}_b(1+\overline{B}_v)-\overline{P}_{rv}-1=0 \quad (23)$$

Considering figure 2 :

- It will be seen that the displacement ratio range is restricted overall to typically $0.75 < \varepsilon < 0.91$ but depends upon the load flow rate to be supplied.
- It will be preferable for the displacement ratio to be closed towards the lower end of the operating range.

Considering figure 3 for the specific load flow rate :

- A displacement ratio beyond 0.882 will not allow the PTU to rotate and power transfer will not exist.
- As the displacement ratio decreases below 0.882 the pressure differential increases and the PTU speed increases. The supply pressure P_1 continually decreases and the pressure to be compensated P_2 increases as required.
- As the displacement ratio decreases further the speed rises to its maximum when the supply pressure P_1 reaches the PRV setting and the equations then do not apply.
- A lower displacement ratio therefore ensures that the driving pressure differential lies beyond the friction dead-band.
- The PTU speed decreases linearly with increasing displacement ratio

Table 1 Data taken on PTUs used in different aircraft [7]

Aircraft	D_m ($10^{-6}m^3/s$)	D_p ($10^{-6}m^3/s$)	Displacement ratio ε	Motor pressure (bar)	Pump pressure (bar)
Uni-directional, fixed displacement					
DC-10/ND-11	1.23	1.13	0.92	200	179
757	4.00	3.65	0.91	172	150
Gulfstream 11	1.73	1.57	0.91	207	200
A-300	4.00	3.46	0.87	207	207
767	0.25	0.21	0.84	112	86
727, 747	0.25	0.21	0.84	207	207
737	0.81	0.63	0.78	169	166
Bi-directional, variable displacement					
DC-10/MD-11	5.00	4.46-5.50	0.89-(1)-1.10	207	193
C-17A	3.15	2.62-3.66	0.83-(1)-1.16	275	255
A-320	2.10	1.57-2.62	0.75-(1)-1.25	207	200

Considering data from aircraft applications, Aerospace recommended practice SAE ARP1280 [7], Table 1 shows actual data for a range of aircraft and for fixed displacement and variable displacement PTUs. The displacement ratio can vary between :

$$0.78<\varepsilon<0.92$$

for uni-directional operation with fixed displacements.

$$0.75<\varepsilon<0.89$$

for bi-directional operation with variable displacements.

$$0.76<\varepsilon<0.88$$

suggested from the theory presented.

For a variable displacement PTU, a swash-plate adjusting control system is required to change the displacement ratio of the in-line axial piston unit depending on the direction of power transfer, figure 4. The displacement ratio would be changed around the neutral position, when no PTU action is needed. Hydromechanical control system is normal practice, and integral with the variable displacement unit, to change swashplate position in the correct direction from neutral.

CONCLUSIONS

The approach presented has a qualitative synergy with the data from typical applications, the linear approximations giving a good first-order insight into PTU behaviour.

An understanding of the interaction between load flow and each pressure relief valve setting is crucial and the resulting design equations are not explicit. However the design only requires solution of quadratic equations.

It is clear that for a particular design the effect of increasing load flow results in a displacement range that increases slightly. It seems preferable to move towards the lower end of the displacement range. The pressure on the healthy side is then closer to its pressure relief valve setting and the speed moves towards a higher value. A good design will try to avoid the extremely high speeds that could occur in practice.

Further work on PTU dynamics is needed, and poses an interesting analysis problem when an actual PTU design is considered for variable displacement units.

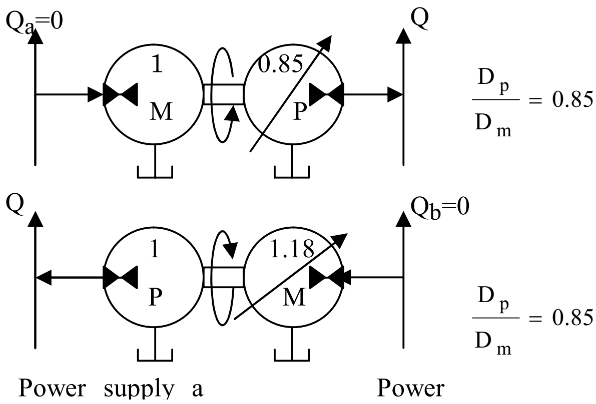

Figure 4 Bi-directional operation of a PTU with displacement

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