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SELF-INDUCED FAULT OF A HYDRAULIC SERVO VALVE

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Keywords: *Self-induced blockage, Hydraulic locking, Servo-valve fault, Brake-by-wire*

Abstract

Two stage hydraulic servo valves equipped with a spool and sleeve assembly and a nozzle and flapper type first stage can fall in a total blockage in a self-induced manner due to fluid dynamic phenomena. The present contribution reports what the working mechanism of this novelty in hydraulic looks like.

Remark

Due to the limited number of pages the present paper is rather a brief report of essential investigation results. A comprehensive description will be released in short.

1. Introduction

An intermittent uncontrollability of hydraulic servo valves equipped with spool & sleeve is called 'hydraulic locking' in technical terminology. This malfunction differs from the mechanical blockage of the spool caused by single or multiple foreign materials or mechanical deformation of a member.

Scientists and engineers have concluded that the spool tends to stick due to increased friction whilst operating. It seems that this type of malfunction occurs more often than that caused by debris. In many fault cases, no debris has been found despite expectation of finding such debris. (cf. Hydraulic Fault at Boeing 737 [1])

The spool of conventional servo valves 'swims' freely inside the sleeve. Its movement will be controlled by the primary stage and return spring. Fig. 1 depicts a principle schematic of such two-stage hydraulic servo valve.

Should the secondary stage be considered alone, the traditional term of 'hydraulic locking' is a bit of a misnomer and overstates the situation, as the spool in the secondary stage sticks only temporarily.

In contrast, a real unrecoverable 'hydraulic lock' occurs in a two stage servo valve when the closed loop control system is no longer able to manage the second stage by means of the primary stage. It is a result of an interaction when both stages mutually influence each other and eventually freeze. Due to this, the closed loop control is no longer able to manage the system. This interaction is more complex than 'sticking' of a spool and sleeve assembly.

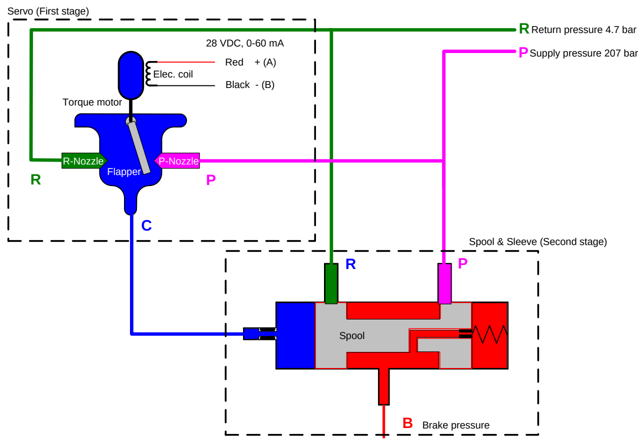


Fig. 1 Principle hydraulic circuit of the investigated two-stage servo valve

This report descripts trouble-shooting results made recently in the field of aircraft engineering and introduces the entire working mechanism of a self-induced fault of a hydraulic servo valve.

2. Occasion for the present investigation

The state-of-the-art brake system of a modern transport aircraft is actuated electro-hydraulically. Such a so-called 'brake-by-wire' system of a regional aircraft, of which manufacturer and brake system provider wish not to be mentioned by names here, failed sporadically whilst operating. The occurrences affected seriously the dispatch ability of the aircraft as one of the duplex brake systems switches off arbitrarily by itself without any typical error patterns. The fault message of the system reads as: “PR MORE THAN BCM COMD”. According to the message the feedback pressure level at the brake cylinder was abnormally higher than the intended set value of the brake control system. In such condition, the control logic switches off the Brake Control Valve (BCV) at the corresponding brake in order to prevent possible damage incurred by overheating.

As the faults scattered almost evenly at all wheels, the troubleshooting was focused on the brake control units. The BCV, which is a pressure controlling, two-stage-hydraulic-servo valve, was eventually under suspicion.

3. Reconstruction of the fault under laboratory condition

During the troubleshooting phase, it was ascertained that there must be a more complex fault with mutual influences between the first and second stages besides the simple fault caused by sticking spool in the second stage. In order to reconstruct the complex phenomenon the BCV was installed on a test bench. The test setup reflects the original A/C flight test instrumentation with which the hydraulic pressure levels are monitored for the supply and return lines as well as for the outlet-port of the brake control valve.

3.1 Experimental investigation Instrumentation / Data Acquisition

As the attention was focused on the pressure development and its variation at whole system members, an extra pressure transducer was installed between the first and second stage (see Fig. 17). The test set-up allowed measuring the actual pressure at all hydraulic lines without causing any influences or disturbances. Data acquisition was conducted at a sampling rate of 200Hz.

3.2 Experiment / Reconstruction Endless repeat of a demanding profile

Although the typical fault condition was more or less known in the advanced stage of the trouble shooting, an exact determination of the working point was not a trivial issue due to the numerous, continuously changing non-linear parameters. Hence, the test was conducted in such a way that a certain command profile was repeated in ‘endless’ manner, and then the supply a/o return pressure, temperature etc. were systematically changed. The command profile consists of a self-test impulse and a terminating command, with which the system performs an in-situ system-check prior to lowering the landing gear in the approach flight phase and decelerates the rest spin of the wheels before retracting the landing gear into the bay after take-off.

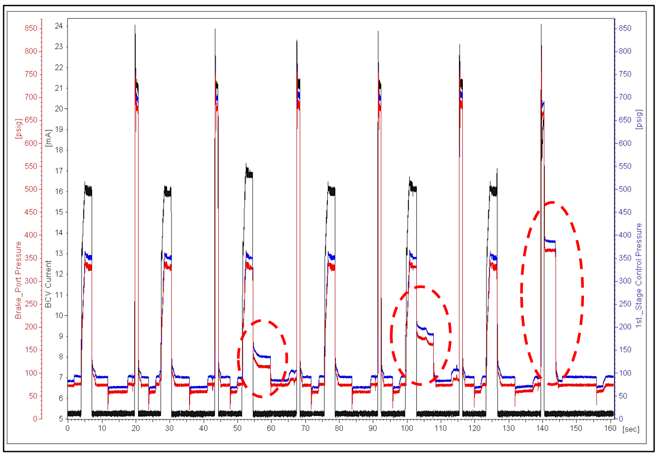


Fig. 2 A selected time record with 3 abnormalities

The test cycle was repeated until a series of faults was eventually registered as shown in Fig. 2. This diagram presents therefore only a small section of the entire test measurement data. The section recorded for approximately 160 seconds contains six system-check impulses called “Pressure Pulse Test” and the same number of so-called “Gear Retract Braking” ramps in change. Note that the valve was switched off at every single impulse or ramp.

For better understanding, the discussions in the following chapters refer to the “GRB (Gear Retract Braking)” and “PPT (Pressure Pulse Test)” with their event numbers. For example, “GRB 1” means the first terminating command started at t = 4 sec, whilst “PPT 6” means the sixth in-situ system-check carried out at t = 139.4 sec. etc.

4. Contemplations of physical background phenomena

Prior to describing the new findings obtained from the present investigation, some fundamental contemplation and observations collected at the beginning of the troubleshooting phase will be discussed for better understandings whilst reading the later chapters.

4.1 Extra force development at the flapper in the first stage

Should the first stage of the servo valve accidentally work across a working point, at which the flow separation occurs, the effect of such a transition is noticeable in the electro-hydraulic signal conversion; by reaching the sub-critical flow region (10^3 < Re < 1.7 ~ 4 10^5), the flow separates on the cylindrical flapper body, so that this experiences an extra drag. The flapper drifts away to the downstream direction as if a sudden suction force would have been developed on its lee side. Whenever such flow separation occurs on the flapper body, the gap at the nozzle increases accordingly a certain amount due to the extra force. The effect of such an extra opening at the nozzle clearly reflects in the pressure answer at the control circuit as shown in Fig. 2 as a sudden increase in the gradient.

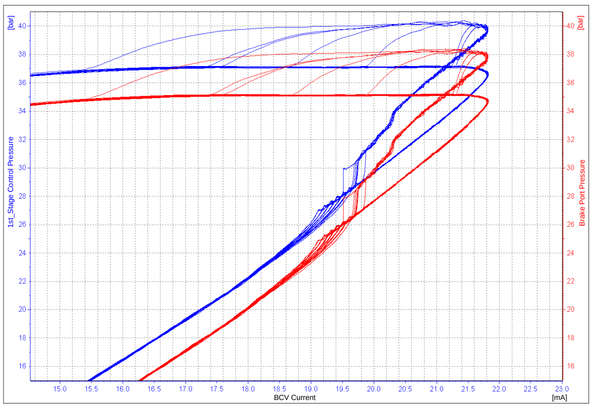


Fig. 2 Sporadic sudden shifting of the pressure response caused by extra opening force at the flapper in the first stage

Fig.3 shows the corresponding time history of the control pressure at the first stage. It is clearly to identify that the pressure answer sporadically inclines to override despite the preset current limitation of 19.8 [mA].

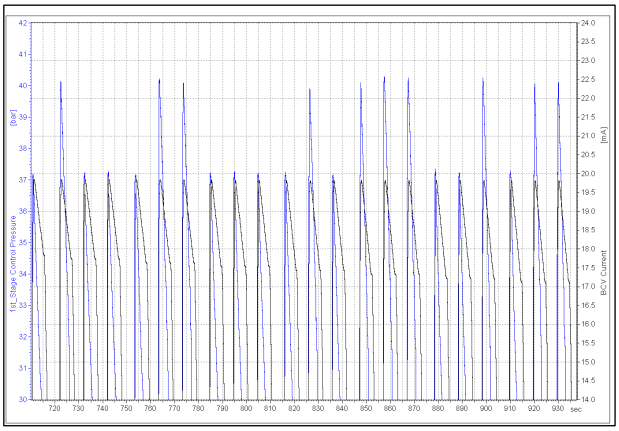


Fig. 3 Sporadic override of the control pressure despite the current limitation

The extra opening of the nozzle will be kept until the reversal of the command - so far the system is able to create a sufficient reset force afterwards to manage the flapper movement.

4.2 Internal leakage and gap tolerance as a significant influence factor

The prerequisite for a properly working second stage is an even gap distance between spool and sleeve throughout the whole circumference of the annular gap. Should the spool be placed eccentrically, the internal leakage and consequently the purging gap stream increases exponentially despite unchanged absolute opening surface area, since the leakage flow is not dependent on the surface opening rate but on the gap amount according to the unsteady two-dimensional Reynolds equation given below [2]:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\eta}\cdot\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\eta}\cdot\frac{\partial p}{\partial z}\right)=6\cdot\frac{\partial}{\partial x}\left(\frac{\partial(h\cdot U)}{\partial x}+\frac{\partial(h\cdot W)}{\partial z}+2\cdot\frac{\partial h}{\partial t}\right)$$

whereas

x, z : axial, tangential coordination, h: gap height, p: pressure, U: wall velocity in x-direction, W: wall velocity in z-direction, η: dynamic viscosity.

The equation points out that the change of the gap height causes an exponential parametric

effect on the internal flow. Fig. 4 shows the geometric condition in the case of a metallic contact due to a non-coaxial misalignment in the sleeve. On the opposite side of the contact point, the gap amount increases to double compared to the original gap distance and consequently the internal leakage, in other words the 'purging quantity', increases abruptly. Once a continuous axial flow is established, the spool is hardly able to recover the original gap by itself because the resulting lateral force of the spool presses the spool to the sleeve wall.
In the case of a radial contact of the spool to the sleeve's wall, it is also to expect that the friction will be drastically increased.

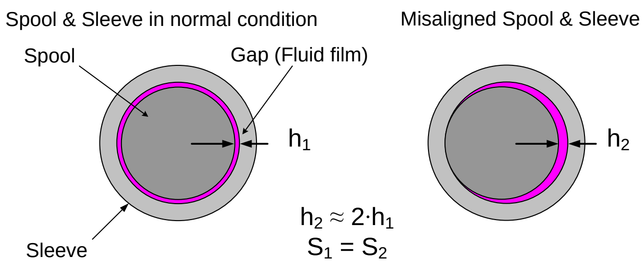


Fig. 4 Change of the gap distance despite same opening surface area

5. Measurement data analysis and interpretations

5.1 Abnormality in the pressure response and interrelations

As described in Chap. 2, the fault becomes noticeable due to abnormal higher pressure, i.e. 'pressure more than commanded'. Fig. 2 shows three abnormalities as such. It must be said that the classification for fault is dependent on the threshold values of the pressure monitoring in the control loop (in most cases both pressure level and dwell).

The first two abnormalities in the record would not be recognized as faults in A/C due to their abnormal but still low pressure level and relative short dwell. In any case, these three abnormalities contributed valuable realizations to the troubleshooting.

Prior to discussing the measurement results in detail, some interrelation facts gained from previous test campaigns will be clarified:

- The pressure response at the brake port is determined by the geometric spool-position and dependent on its actual running speed. According to the character tests the pressure behavior is initially at static condition quite linear then non-linear and increasingly hysteretic at enlarged demanding speed.
- The gradient in the brake pressure curve reflects the actual running speed of the spool.
- As 'communicating vessels' the ports and their associated components have mutual influences.
- The power consumption of the torque motor and consequently the current measured at the magnetic coil depend on the actual load applied to the hydro-mechanical part of the first stage of the valve.

5.2 Stability of the pressure signal answer and mobility of the spool

Comparing the fault cases with normal cycles, it is striking that there are significant differences in pressure variations at the brake port. (cf. Fig.5 vs Fig. 6).

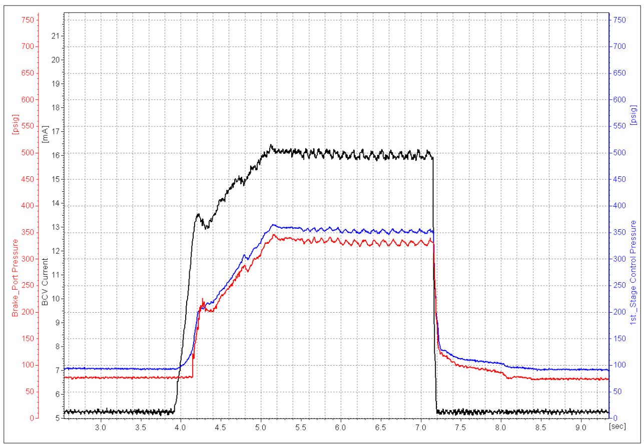


Fig. 5 Dithering during the GRB 1; normal / no fault

Considering that the pressure variation at a certain pressure level reflects nothing but a small dithering of the spool in its actual position, it seems that the spool must be kept in motion in order not to fail: In all three abnormal cases, the motion of the spool becomes 'hyper-stable' before the "Pressure more than

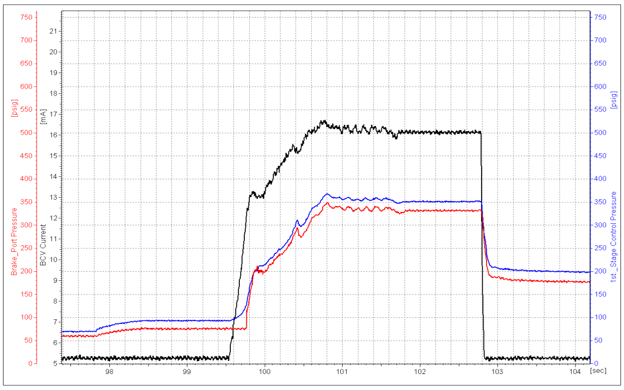


Fig. 6 Stop dithering during GRB 5; abnormal

commanded" event occurs. As long as the spool remains 'nervous', the valve works fine. There are even cases at which the spool recovers its dithering after having been 'hyper-stable' (cf. Fig. 7). The system was end up not faulty.

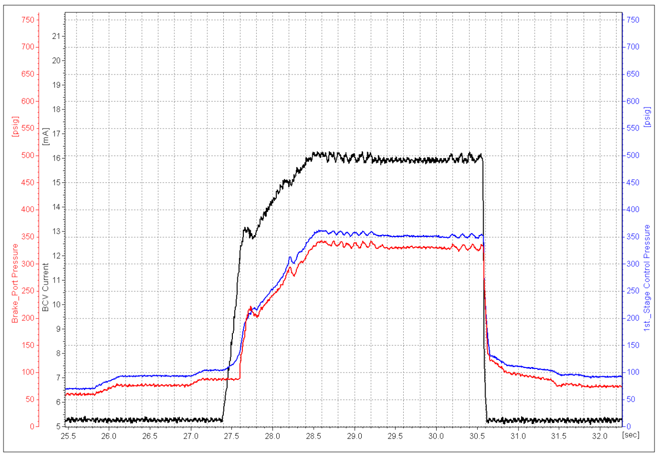


Fig. 7 Dithering recovery during the GRB 2; normal / no fault

The spool showed exactly the same faulty behavior whilst PPT. It must be said that the mobility of the spool at a given command level depends mainly on the friction.

This is the first direct indication that the fault must be a mechanical problem or at least initialized by one or more mechanical parametric disturbances.

5.3 Switching-off dynamic of the spool

Considering that the viscosity and the bulk modulus are constant within a short time interval, the switching-off dynamic of the spool must be quite similar whenever the valve is switched off at a more or less equal spool position. Such changes in characteristic diagrams, however, are only recognizable when an adequate scale is chosen. cf., Fig. 5 & Fig. 6 vs. Fig. 8 & Fig 9.

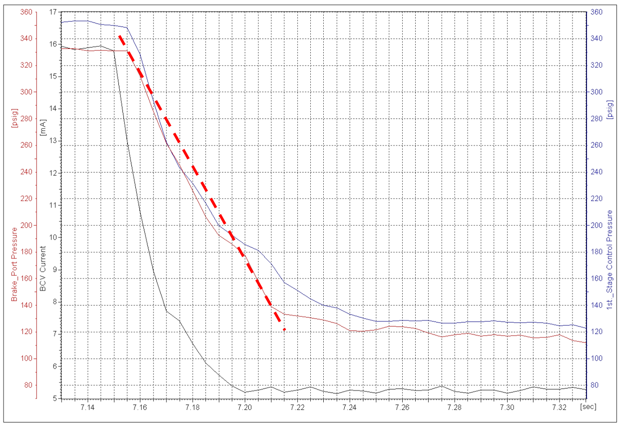


Fig. 8 Steep gradient at Switching-off: GRB 1; normal / no fault

The pressure plots of faulty cases show a significant difference in the gradient of their curves compared to those of corresponding normal cases. Fig. 8 and Fig. 9 show representative the difference in the case of GRB, whilst Fig. 10 shows such for PPT.

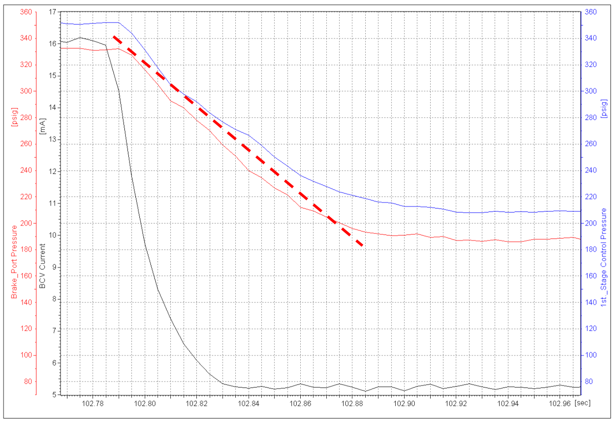
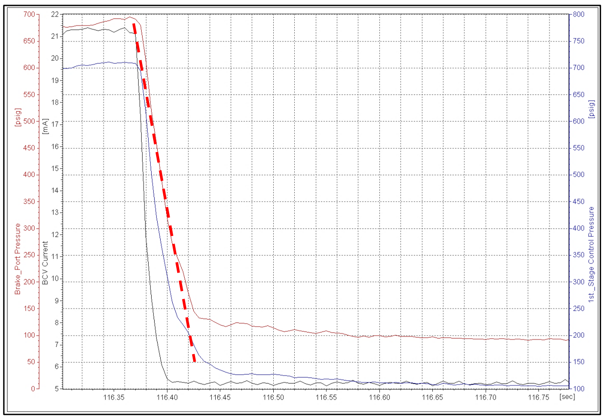


Fig. 9 Gentle gradient at Switching-off: GRB 5; abnormal

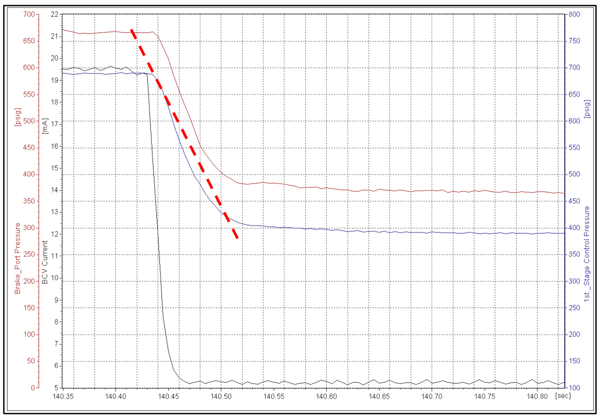
Note that the gradients of the brake-pressure curve at the switching-off phase are different in the cases of PPT and GRB because of the different operating loads at the bias-spring.

In the case of normal operation, the signal response at the 'switch off', i.e. the pressure dumping at the brake port is rapid and similar under the same command type.

Despite the very same switch-off condition, the pressure dumping process shows significant differences in the fault cases. The movement of the spool seems to become more sluggish since the gradient of the curve reflects in the first instance the running speed of the spool. Again, the reason for changing in the running speed of the spool can only be the change of the actual friction between the spool and the sleeve.



Steep gradient at PPT 5; normal / no fault



Gentle gradient at PPT 6; abnormal

Fig. 10 Different Gradients at Switching-off

5.4 Feedback intensity in the water hammer effect

The feedback pressure level at the return line offers an extra valuable indication to corroborate the interpretation of friction changing in the second stage.

	RPmax [psig]	at t = [sec]		RPmax [psig]	at t = [sec]
GRB1	147.55	7.180	PPT1	215.78	20.845
GRB2	150.83	30.595	PPT2	210.16	44.410
GRB3	**137.40**	54.600	PPT3	219.06	68.465
GRB4	148.17	78.810	PPT4	200.48	92.425
GRB5	**139.12**	102.820	PPT5	205.48	116.390
GRB6	146.77	126.625	**PPT6**	**125.38**	140.460

Tab. 1 Return Pressure Peaks at switching off

The pressure impulse caused by a radical conversion of kinetic energy into static pressure, known as 'water hammer effect', shows a significant difference in the fault cases:
In the case of a normal operation, a proper water hammer effect is to be found. In the fault cases, however, it is far less intensive due to the unintended but gentle closing of the brake port. Peaks of the return pressure listed in Tab. 1 show the significant difference in the case of fault highlighted by red.

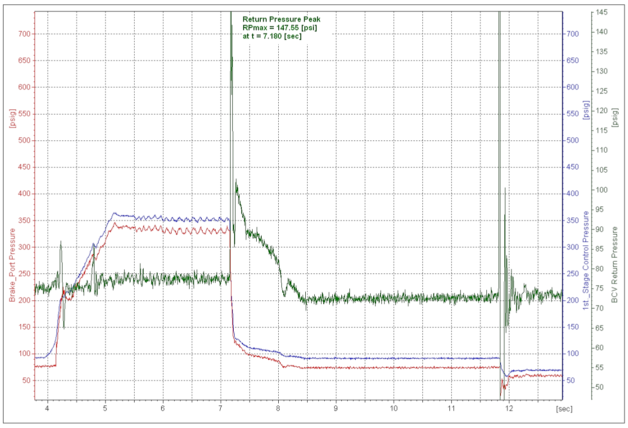


Fig. 11 Full water hammer effect at the switching-off: GRB 1: normal / no fault

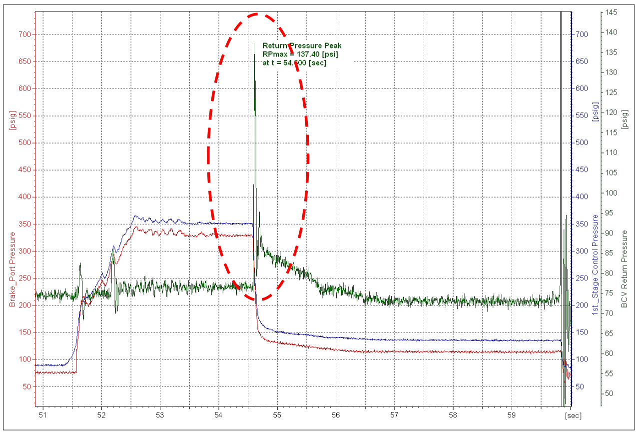


Fig. 12 Reduced water hammer effect at the switching-off: GRB 3: abnormal

A discrete changing from 'radical and abrupt' to 'slow and gentle' manner in closing of the port is to be concluded as an effect of reduced speed of the spool and consequently as a result of increased friction between the spool and the sleeve as long as the reset of the spool is managed only by a bias spring.

Fig. 11 vs Fig. 12 and Fig. 13 vs Fig. 14 show pairwise the clear differences of the return pressure peak between the normal and fault cases after the switching off the electric coil.

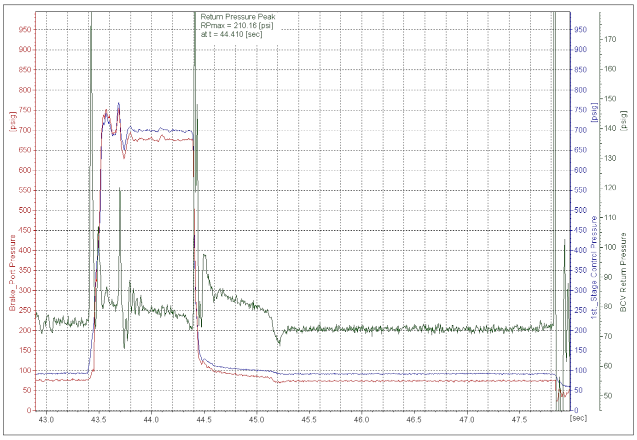


Fig. 13 Full water hammer effect at the switching-off: PPT 2: normal / no fault

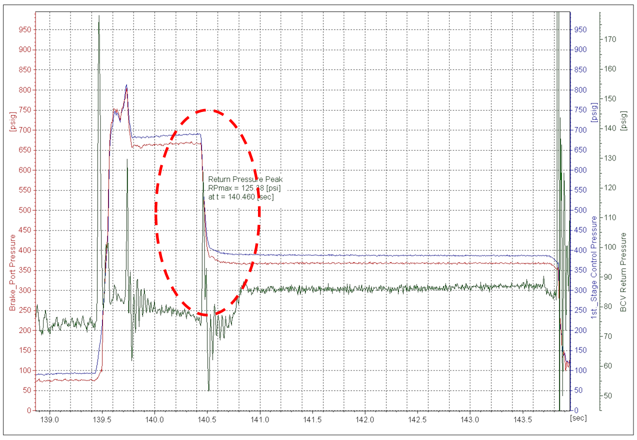


Fig. 14 Reduced water hammer effect at the switching-off: PPT 6: abnormal / fault

5.5 Fluid dumping behavior at the control circuit

In normal cases, whenever the solenoid is switched off and accordingly the flapper closes the inlet nozzle due to the spring preload, then the control circuit dumps the fluid trapped in the spool's control chamber via the outlet nozzle as the coil-spring resets the spool back to the start position. (cf. Fig. 1)

Such dumping processes are traceable in the record of the return pressure. For better understanding, the following description refers to the Fig. 15 and Fig. 16, of which time frames are from 19.0 sec to 24.0 sec, from 139.0 sec to 143.3 sec, respectively. It must be said that this system behavior is generally quite reproducible, regardless of the ramp profile of the commands.

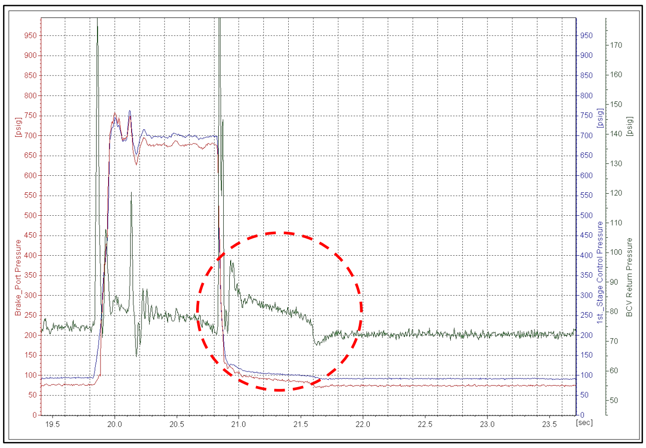


Fig. 15 Return pressure development after PPT 1: normal / no fault

The dumping process seems to take 800 ± 100 [ms] depending on the last pressure level in the control chamber. Completing the process, the return line pressure stabilizes to a slightly lower level compared to that adjusted once under the operational condition (approximately 78 psi).

Note that the outlet nozzle drains continuously a small amount of fluid whilst the solenoid is demanded. This is not leakage but 'regular fluid consumption'. This is the reason why the return pressure level sinks slightly below when the flapper is completely closed (72 psi at t ≥ 21.8 sec / Fig. 15).

In the case of fault, however, the return pressure stabilizes in an increased pressure level (approximately 86 psi at t ≥ 140.9 sec / Fig 16). This is definitely a sign that the inlet nozzle is not yet completely closed in spite of the switched off solenoid.

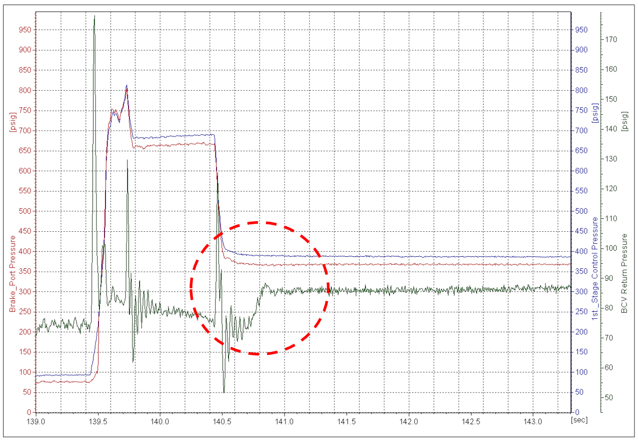


Fig. 16 Return pressure development after PPT 6: abnormal / fault

It is striking also that the return pressure sinks down right after the switching off temporarily to the lowest possible level (for the present case approximately 75 psi at t ≈ 140.6 sec / Fig 16) before the return pressure increases to approximately 86 psi and stabilizes there. The temporal reduction of the return pressure means that the flow resistance (i.e. Lohm value) between the pressure transducer PR and the inlet nozzle, i.e. downward from the source (see Fig. 17), has been significantly reduced. The only possible reason for such a sudden reduction of the flow resistance is an extra drain path due to a misalignment of the spool in the sleeve (i.e. non-coaxial alignment of the spool. cf. Chap. 4.2).

In the normal case shown in Fig. 15 the water hammer effect disturbs the pressure dumping process. The bias spring in the second stage overcomes the disturbance and continues the process. The changing in the curve’s gradient at t = 21.06 sec reflects this. The buckle, however, is only a result of the superposition of the pressure wave created by the water hammer effect and the dumping pressure created by the bias spring. A damping effect against the direction changing of the pressure wave is recognized at t = 21.03 sec. The dumping process seems to be continued until t = 21.63 sec.

No matter what kind of mutual interferences ever happen between the first and second stage, the first stage has in the normal case only one drain, which is the regular outlet nozzle (R-nozzle). And the flapper is able to close reliably the inlet nozzle (P-nozzle).

In the fault case shown in Fig. 16, however, the flapper seems not to have a chance to close the inlet nozzle. Taking a new extra drain route, i.e. an additional connection to the return line via the gap between the spool and sleeve wall (cf. Chap. 4.2), the inlet nozzle might have developed a constant internal flow as a result of the dramatically increased internal leakage in the spool and sleeve assembly (cf. t ≥ 140.9 sec / Fig. 16).

5.6 Hegemony loss of the first stage, Completion of the self-induction

According to the plots shown in Fig. 16 the return pressure sinks down right after the switching off to approx. 75 psi before it increases again to approx. 86 psi and stabilizes there. As discussed in Chap. 5.5, this is the evidence that the inlet nozzle in the first stage is still open because the return pressure does not sink down further to approx. 72 psi level. Hence, an internal flow must have been developed strong enough across the first and second stages to keep the inlet nozzle in an open position. In other words; the suction force at the flapper is so high that the flapper is no longer able to close the inlet nozzle by itself. It seems that the flow around the flapper separates the cylindrical flapper body during the process (140.7 ≤ t ≤ 140.9) by which the flapper experiences an extra amount of suction force. (cf. Chap. 4.1) Having an extra amount of force, the resulting vector sum might be strong enough to overcome the closing spring force of the flapper or even to demand a further opening. The first stage loses definitively the hegemony in any case due to the fully developed, steady turbulent flow around the flapper.

5.7 Development of a new drainage and the loss of command ability

Considering the coherency between the internal leakage and effective gap distance discussed in Chap. 4.2, it is clear that the first stage drains in the case of spool’s misalignment its incoming fluid suddenly in two routes.

Fig. 17 depicts the situation schematically. The

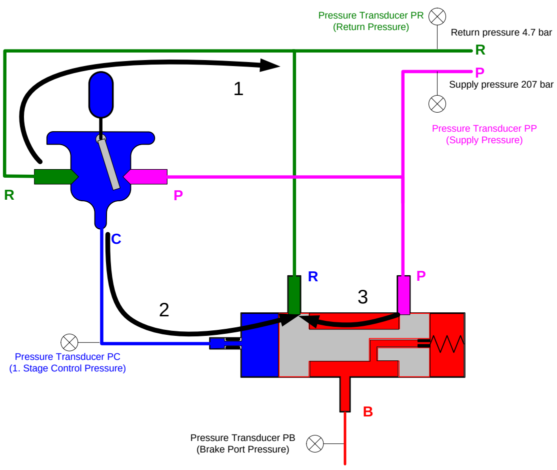


Fig. 17 Drain routes of the hydraulic fluid in the case of the fault

channel between the first stage and the second stage, i.e. the route No. 2 in the figure is no longer a bidirectional command line. This channel now works parallel to the original outlet channel of the first stage, i.e. the route No. 1, as a new unidirectional drainage. The severe implication is that the flapper is no longer manageable due to the high flow force at the inlet nozzle.

6. Reconstruction of the entire fault working mechanism

Implications based on the physical phenomena discussed in Chap. 4, as well as the data analysis and interpretations described in Chap. 5, allow a reconstruction of what the root cause of a self-induced hydraulic locking looks like:

When the spool in the second stage of a hydraulic servo valve becomes sluggish due to increased internal friction, the orientation of the spool could be occasionally aligned no longer coaxial relative to the sleeve. If such a misalignment occurs, the effective gap distance between the spool and sleeve can increase up to 100%. Accordingly, the internal leakage increases abruptly and a constant flow develops in axial direction among the spool and sleeve gap. Then, the first stage dumps no longer solely via the outlet nozzle but also via the communicating vessel, i.e. through the control line additionally into the return line. The flow rate at the inlet nozzle increases dramatically since the sink pressure at the opposite side has been decreased. The arising effect is that the second stage seriously affects the controllability of the first stage as the flapper is hardly able to close the inlet nozzle. At a certain working point where the flow separating point on the flapper's lee side wanders in an upstream direction, the increased flow rate amplifies therefore the resulting flow force. Hence, the opening rate at the inlet nozzle increases. This again increases the total internal leakage / drain at the second stage. Once such a mutual influence is initialized, the escalation effect occurs so rapidly that the system does not have any chance to recover its controllability and eventually freezes by itself.

7. Discussion - Arbitrariness of the fault Reason for randomly happening events

The arising question is now why the self-induced hydraulic locking happens randomly so that the fault is not always to be observed in spite of unchanged working conditions. In this chapter the reason for the random scattering will be discussed.

The random characteristic very much resembles 'Russian roulette' as the actual angular position of a rolling, non-perfect cylindrical body in a likewise non-perfect borehole will be randomly determined by hydraulic flow which passes the cross section of the assembly. Consequently its amount is hardly predictable. Furthermore, during operation, a quasi-axial stream will be determined by unpredictable internal leakage at a given coincidental position. The stream in axial direction can be either aiding or inhibiting. This additional fact makes the prediction of spool's actual angular position fully impossible.
Previous experiments showed that the running speed of the spool changes and it is dependent on the actual angular position of the spool relative to the sleeve. Schlemmer et. al. [2] carried out a similar investigation and reported a certain angle dependency of the running friction of the spool.
Moreover, it is recognized that the sliding ability at the same point is getting worse with increasing numbers of cycles. The reason for such changing in sliding ability seems to be the degeneration of the lubrication film caused by rubbing..

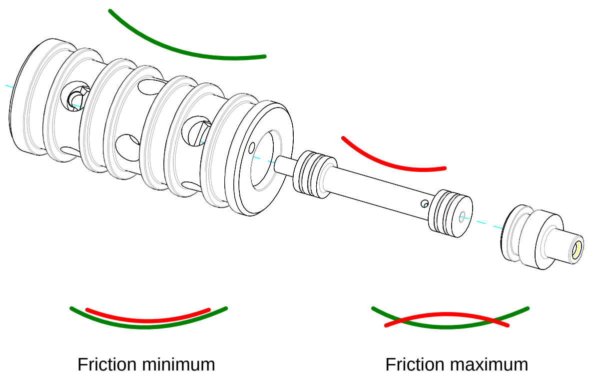


Fig. 18 Determination of the actual friction

Even though the susceptibility of an angular section caused by the misfortunately aligned parts could be identified in advance, the fault occurs arbitrarily as long as the inevitable geometric misalignment inclines to play a dominant role.

8. Conclusion

A two stage hydraulic control valve consisting of an electro-magnetic servo valve of flapper type and a spool and sleeve assembly can lose its controllability completely if both stages influence each other mutually. Becoming sluggish due to direct contact of the spool to the sleeve wall, the second stage initializes the fault and finally escalates the situation by starting the process of a high internal leakage caused by misalignment of the spool in the sleeve. Once the self-induction commences, the process is irreversible as soon as the first stage loses its controllability due to the high flow rate. Should the flow force caused by internal leakage not be high enough, so that the electric coil and the return spring of the flapper are still able to manage the movement at the spool by themselves or they even overcome the blockage by some chance, e.g. vibration or jolting, then the system recovers its controllability. Whenever the control loop still manages the servo valve within the predefined threshold of time-out, the fault can be masked and remain undetected.

9. Summary

Two stage hydraulic servo valves equipped with a spool and sleeve assembly and a nozzle and flapper type first stage can fall in a total blockage due to fluid dynamic phenomena.

The initialization happens at the second stage in the manner of 'Russian roulette' principle. Moving in a non-perfectly straight and non-ideally round borehole, the likewise non-perfect spool can coincidently misalign itself due to actual insufficient hydro-static lubrication condition. As soon as the spool stands no longer co-axially, the internal leakage increases exponentially due to the viscous hydraulic fluid as working medium.

After being initialized by such a mechanical misfortune, the internal leakage flow can develop between the spool and the sleeve so critically that the flapper in the first stage can barely manage the inlet nozzle despite the support of the preloaded spring force. In the worst case the flow separates behind the cylindrical flapper body. Due to the increased drag the force balance is so seriously disturbed that the flapper drifts away from its position as if a sudden extra opening force has been developed. Then, the servo valve is eventually no longer manageable by electric command signals. Once induced by the second stage to the fault, the first stage stabilizes the state against disturbances since the flow condition around the cylindrical flapper varies/shuttles between sub-critical and super-critical Reynolds number regions.

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