Here is my understanding of the system and the experiments:

The considered system contains a cylinder subjected to some gravitational load. The gravitational load changes somewhat with the piston travel, indicating that the cylinder is connected to some kind of boom. Further, there is a characteristic jump in piston side pressure whenever the direction of motion changes corresponding to a certain amount of friction.
Cylinder data: Piston diameter, $d = 100\,mm$, rod diameter, $d_R = 56\,mm$. This corresponds to a piston area of $A = 7854\,cm^2$, an annulus area of $A_A = 5391\,cm^2$, and a cylinder area ratio of $\varphi = \frac{A_A}{A} = 0.6864$.

The cylinder is connected to a PVG32 with a 40 l/min spool, $Q_{max} = 40\,\frac{l}{min}$. The spool has a linear flow characteristic; it is symmetrical and closed in neutral. The electrohydraulic actuation corresponds to a 12V valve, where 6V nominally is neutral and 3V and 9V, respectively, represent maximum spool travel to either side.
The main spool of a PVG32 travels $\pm x_{max} = \pm 7\,mm$ and for a linear spool the deadband is $x_0 \pm 0.8\,mm$. Introducing a dimensionless spool travel parameter $u = \frac{x}{x_{max}}$ we have the following, nominal characteristics for the metering-in flow:

$$Q_{in}(u) = \begin{cases} \dfrac{u-u_0}{1-u_0}\cdot Q_{max} & u \geq u_0 \\ 0 & u_0 \geq u \geq -u_0 \\ \dfrac{-u-u_0}{1-u_0}\cdot Q_{max} & -u_0 \geq u \end{cases} \qquad (1)$$

Nominal behavior requires 4 things:

1. that the electrohydraulic actuation is perfectly centered and symmetrical,
2. that there is sufficient pump flow to deliver the desired metering-in flow,
3. that the pump pressure is at least $p_R = 7bar$ higher than the metering-in pressure,
4. that there is sufficient back-pressure on the metering-out side to avoid cavitation.

In (1) the metering-in flow is always positive and the dimensionless deadband is then $u_0 = \frac{0.8}{7} = 0.1143$. The dimensionless travel can be linked to the signal voltage, U_s, as:

$$u = \frac{U_N - U_s}{3} \qquad (2)$$

This yields a positive dimensionless travel if $U_s < U_N$ and in our case this corresponds to metering-in to the A-port = piston side of the cylinder. For the nominal situation we have $U_N = 6V$.

For a typical cycle (extending and then retracting the cylinder) we have, approximately, the signal voltage shown in Figure 1.

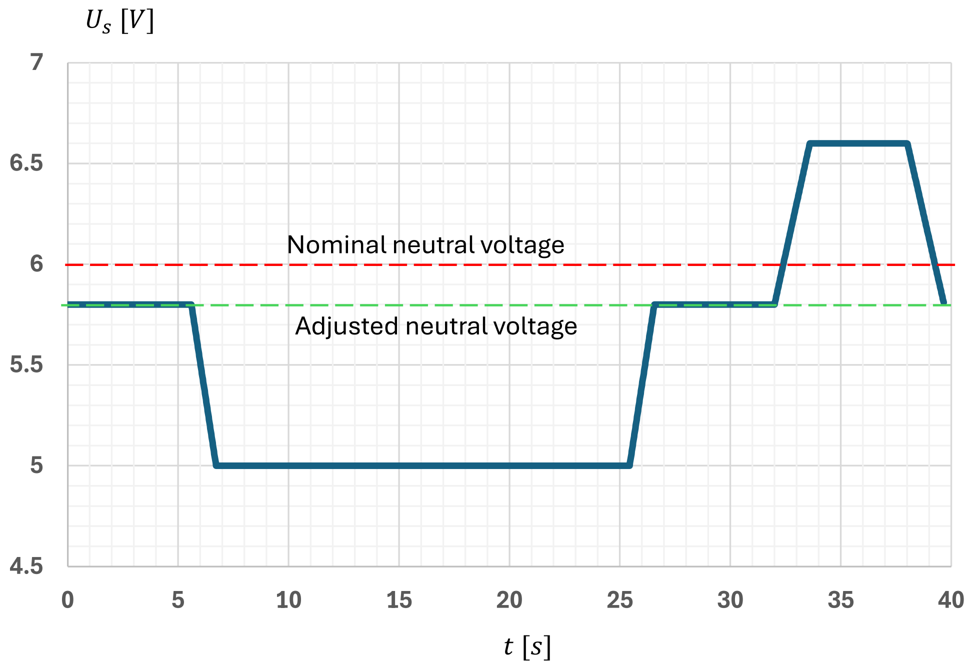

Figure 1. Signal voltage vs. time for a typical cycle.

If we use this data directly together with the logged input voltage we get the following velocities that we can compare with a cylinder velocity derived from the measured cylinder position, see Figure 2.

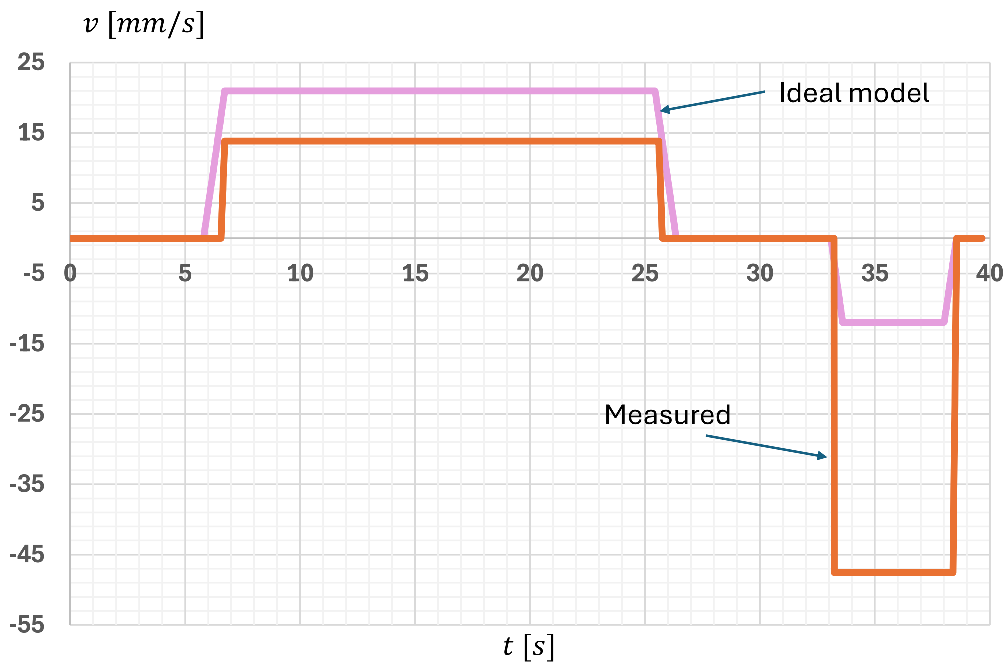

Figure 2. Piston velocities vs. time for a typical cycle. Velocity based on ideal behavior and velocity derived from the measured piston position are shown.

Clearly, we do not have ideal behavior in neither the extracting nor the retracting phase. If we consider the extracting phase, then it seems that (as we often see) the deadband has gone off-center and that the electrohydraulic actuation has a neutral voltage of $5.8V$. This adjusted neutral voltage is shown in Figure 1 and I assume that it has been identified by trial-and-error since it is used as initial voltage. With the adjusted neutral voltage, $U_N = 5.8V$, we can compare the velocities again, see Figure 3.

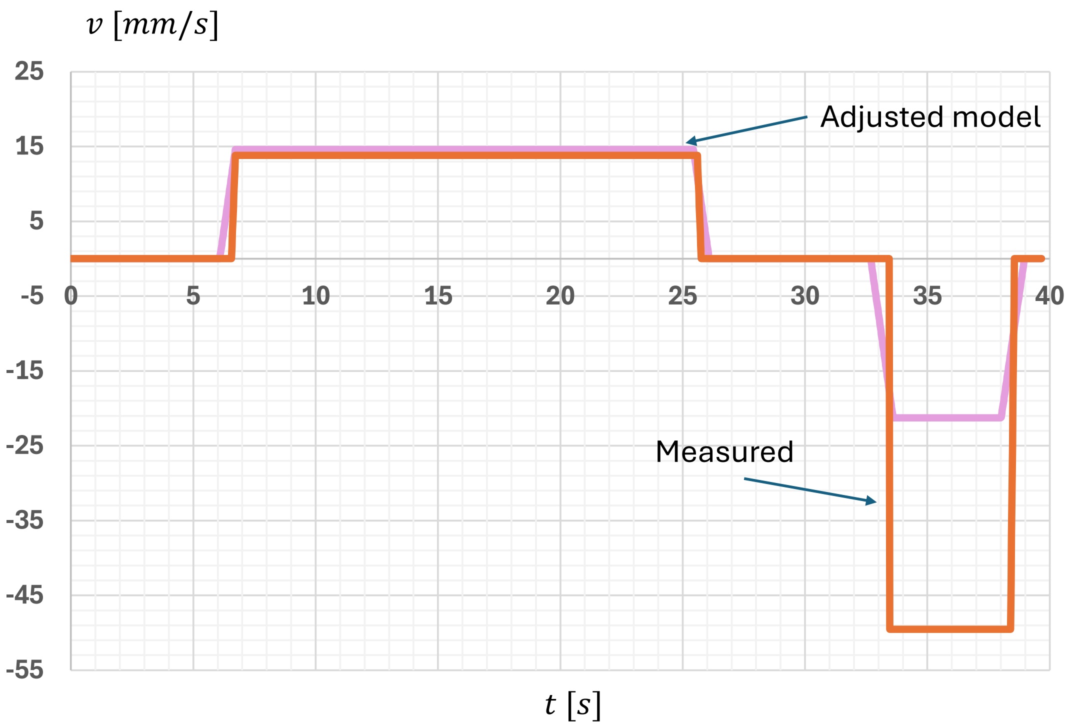

Figure 3. Piston velocities vs. time for a typical cycle. Velocity based on an adjusted neutral voltage and velocity derived from the measured piston position are shown.

This looks much better when we consider the extracting phase and it seems that the other 3 conditions for pressure compensated behavior are fulfilled.
Clearly, this is not the case for the retraction phase. During retraction we meter-in flow to the B-side and from the measurements it is clear that the B-port pressure either cavitates or is very close to cavitating during the entire retraction phase. This is supported by the fact that there is no counterbalance valve or similar load holding/brake valve in the system. Therefore, the PVG32 cannot deliver pressure compensated flow and Equation (1) is simply not valid anymore.
So what does actually control the lowering speed when the pressure compensation is abandoned? Since the B-port either cavitates and/or gets extra oil from some suction valve (?) then the necessary high pressure in the A-port is generated via throttling across the A-T connection. My guess is that the lowering has been adjusted by trial-and-error so that an appropriately small opening (A-T orifice) creates the necessary piston-side pressure via throttling for a reasonable retraction velocity.

To investigate this, we can look at the valve constant. Since the valve is symmetrical and rated to $Q_{max} = 40\ \frac{l}{min}$ when fully opened at a pressure drop of $p_R = 7bar$ we can compute the maximum valve constant from:

$$Q_{max} = K_{v,max} \cdot \sqrt{p_R} \Longrightarrow K_{v,max} = \frac{Q_{max}}{\sqrt{p_R}} = \frac{40}{\sqrt{7}} = 15.119\ \frac{l}{min \cdot \sqrt{bar}} \qquad (3)$$

The maximum valve constant can be used for any situation (flow, Q, opening, u, and pressure drop, Δp) as:

$$Q = K_{v,max} \cdot u \cdot \sqrt{\Delta p} \qquad (4)$$

Since we measure the pressure in the A-port, see Figure 4, and always have an estimate of u, we can simply computed the speed during retraction as:

$$v = -\frac{Q_{A \to T}}{A} = \frac{K_{v,max} \cdot u \cdot \sqrt{p_A}}{A}$$	(5)

Obviously, this is associated with several uncertainties, but the results, see Figure 5, that this could be a good estimate of what is happening during retraction.

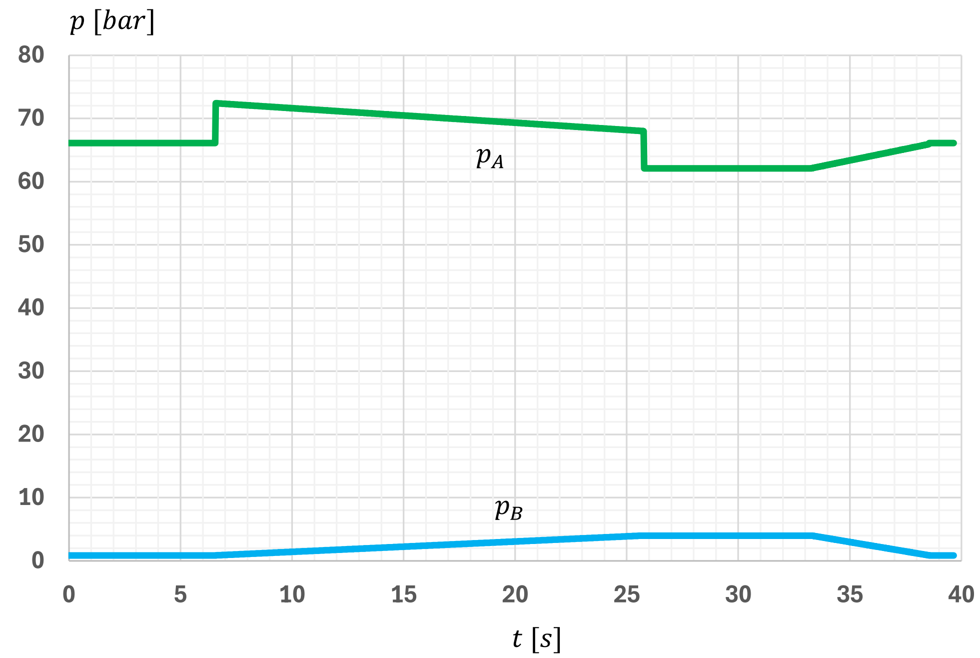

Figure 4. Pressures in the A- and B-port of the system. High frequency oscillations have been filtered out and nonlinear variations have been simplified (linearized).

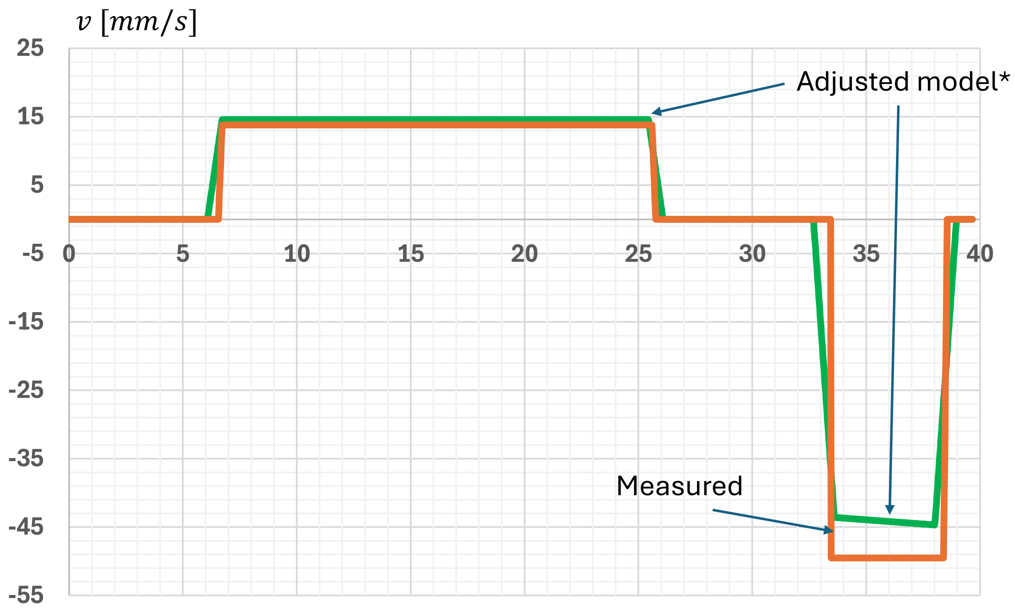

Figure 5. Piston velocities vs. time for a typical cycle. The modelled velocity is based on pressure compensation with an adjusted (off-center) deadband and loss of pressure compensation during retraction.

My assumptions/conclusions:

- **The PVG32 works as expected during extraction, and from a mechanical point of view, we have a kinematic driver delivering a velocity directly proportional to the input signal.**
- **During retraction we lose that control, and the velocity now depends both on the control signal and the piston side pressure.**
- **The dynamic variations of the pressures are, in general, associated with the flexibility of the mechanical system and the liquid compressibility together with the cylinder friction.**
- **The extremely low B-port pressure is closely associated with the cavitation or near-cavitation conditions during retraction. Very low pressures will reduce the bulk modulus (stiffness) of the liquid to near zero and heavily delay any pressure built up.**

The Matlab code I've used to set up the different steady-state characteristics are divided into two scripts:

"`PVG32_Model_Flow_Basic.m`" (ideal/nominal behavior)

"`PVG32_Model_Flow_AssymetricDB_Cav.m`" (off center deadband + cavitation during retraction)