USE OF BEBC 1m MODEL FOR BELLows TESTING.

WORST CASE CONDITIONS FOR THE EXPANSION END STOPS AND THE CHAMBER PRESSURE.

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1. INTRODUCTION

The use, in RCBC, of a GRP bellows to seal the expansion piston to the chamber body, imposes stringent and unusual requirements to the design of the end stops which are placed inside the expansion motor to limit the displacement in both directions. All is stemming from the fact that, using a nearly full diameter piston, the needed stroke is small (6 to 7 mm) and consequently the bellows is designed for rather small excursions from the neutral position (∓ 7 mm)\(^1\). Allowing for 1 mm margin in the available stroke and 2 mm for overshooting at the end of the recompression period (thus obtaining the shortest expansion cycle), we see on Fig.1 that the needed free zone of displacement is 10 mm. If the total movement is to be limited at 14 mm for the bellows, only 2 mm are available on each side to decelerate the moving mass and bring it to a nice stop without creating dangerous forces on the piston assembly.

The same analysis is valid for the testing of the GRP bellows inside BEBC 1m model and new end stops have to be mounted inside the "maquette" expansion motor. In addition to protect the bellows during the tests, this will also give the opportunity of testing the principle of these end stops before the completion of the design of RCBC expansion motor.

2. WORST CASE CONDITION FOR THE END STOPS

To design the end stops for the "maquette" expansion motor, the worst case condition was defined as follows:
a) The modulus of compressibility of the fluid on the chamber side is supposed to be so low that no force is created by the cold piston displacement. This is the case, for example, when a large volume of gas is present together with the liquid.

b) The moving mass is accelerated from its rest position using the full power of the motor.

In these conditions, the 200 kg moving mass arrives against the end stop at a speed of 2.3 m/s.

Nota: The speed of 2.3 m/s is the estimated maximum speed attainable after a displacement of 10 mm. It must be noted that if the displacement was not limited the maximum speed attainable using the full flow of the valves under 210 bar pressure drop would be 4.6 m/s.

3. END STOPS DESIGN

The chosen solution consists of placing on each side of the motor piston ten $\phi 21$ mm hydraulic plungers (having a stroke of 2 mm) which displacements force oil out of the 1136 mm$^3$ backing volumes through calibrated holes Fig.2. Each plunger is forced in the stand-by position by a stack of 6 belleville washers (which provides a force of 220 N in the stand-by position and 1200 N after a stroke of 2 mm).

For each backing volume there are two calibrated holes, a $\phi 0.9$ mm orifice in the plunger center and a $4 \times 0.2$ mm$^2$ slot in the cylinder wall. The last one is so placed that the plunger blocks it up after only 1.4 mm displacement leaving only the $\phi 0.9$ mm hole for the oil to escape. Fig.3 shows that such a geometry brings the moving mass to rest after 1.7 mm. The mean deceleration is 1860 m/s$^2$ (186 g) which is acceptable for the cold piston assembly [2]. The peak pressure in the cavities reaches 1100 bar.

It is clear that air must not be present inside the backing volumes for the system to function properly. After each emptying of the hydraulic circuit, the plungers will be manoeuvred several times at the start-up, using the hydraulic piston to force air out of the cavities.
4. **WORST CASE CONDITIONS FOR THE "MAQUETTE" PRESSURE**

It must be noted that the "maquette" expansion motor is not designed to support a large offset static force. The choice of the bellows backing pressure will thus be limited if one wants to approach a 10 ms expansion cycle. For this worst case conditions analysis this backing pressure is supposed to be equal to the chamber pressure. Also the chamber being filled with a mixture of liquid nitrogen and helium gas to simulate the compressibility of liquid hydrogen, the modulus of compressibility of the chamber fluid is bound to vary from the optimal value. This being said 2 different worst case conditions may be defined.

1) The chamber static pressure is set when the hydraulic piston is in the "before expansion position" (4 mm from the upper limit) and the modulus of compressibility is right (a 7 mm stroke gives a 6 bar \( \Delta P \)).

   In these conditions, if the piston is forced toward the upper stop the pressure on the chamber side is given by the available stroke (i.e. 4 mm). If the chamber static pressure is 7 bar, the pressure will reach \( 7 + 4 \times \frac{6}{7} = 10.4 \) bar.

2) Either the static pressure is set when the hydraulic piston is in a lower position than the "before expansion position" or the modulus of compressibility of the chamber fluid is higher than the modulus of hydrogen (lack of helium gas for example).

   In these conditions, if the piston is forced toward the upper stop, the pressure in the chamber is given by the hydraulic force available. The respective area of the cold and hydraulic pistons being respectively 4400 cm\(^2\) and 100 cm\(^2\) and the hydraulic pressure being 210 bar, the equilibrium is reached for an over-pressure of \( 210 \times \frac{100}{4400} = 4.50 \) bar on the chamber side. The pressure will thus reach

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7 + 4.50 = 11.50 \text{ bar}.
\]
Based on a previous experience, it is expected that the probability of such mishaps will not exceed one per half million expansions.

If the bellows backing gas pressure may be lowered by 1 or 2 bar the peak pressure will be lowered by the same amount.

Nota: It is possible that the bellows cannot be cycled under the conditions described in the previous paragraph. Because, if the backing gas pressure may not be lowered (to maintain the speed) below the chamber pressure, the bellows would be submitted to a varying differential pressure of 0 to 6 bar (external) for a displacement from 0 to 7 mm. This is too much as the bellows is designed for infinite fatigue life when submitted to a varying differential pressure of -3 to 3 bar for a displacement from 0 to 7 mm [1].

In this case, the only way out will be to set the chamber pressure and the bellows backing gas pressure to 4 bar when the moving gear is in the bellows neutral position (3.5 mm) and to impose a sine expansion (Fig. 4). Although generally less severe this test should nevertheless furnish good indications on the bellows fatigue behaviour.

In all the cases, the maximum external buckling pressure which could be applied to the bellows by the expansion motor is limited by the available force to: $210 \times \frac{100}{4400} = 4.5$ bar (4.2) which is acceptable [2].

REFERENCES

[1] Note Rutherford Laboratory EHS/RCBC/R/T/034/BRD.

IDENTIFICATION OF THE VARIOUS ZONES INSIDE MAQUETTE MOTOR

FIG: 1
PLUNGER FOR END STOPS IN MAQUETTE MOTOR

FIG: 2
PERFORMANCE OF NEW END STOPS IN MAQUETTE EXPANSION SYSTEM

fig 3
POSSIBLE EXPANSION SHAPES FOR BELLOWS TESTING

FIG: 4