In order to maintain equilibrium in our free body, intuitively one can see that either L_1 & L_2 must increase and/or P_1, P_2, P_3 & P_4 must decrease in some proportion to the magnitude of F_V.

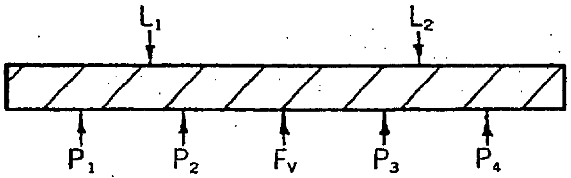

FIG. 2

Considering the elasticity of the bolts,

$$L_1 = L_2 = k_B \Delta \ell = L_B,$$

where k_B is the spring constant of the bolts and $\Delta \ell$ is the incremental change in bolt length. Further, the gasket being elastic (ideally) has a similar characteristic:

$$\sum_{n=1}^{4} P_n = k_G \Delta G = \sum P_G,$$

where k_G is the spring rate of the gasket and ΔG is the incremental change in gasket thickness.

From the foregoing, we know that

$$F_V + \sum P_G = \sum L_B,$$

or,

$$F_V = \sum L_B - \sum P_G$$

Thus, when $F_V = 0$, or approximately so,

$$\sum L_B = \sum P_G \quad ;$$

but, when F_V is at its maximum, $\sum L_B$ will be at its maximum and $\sum P_G$ will be at its minimum. The precise magnitudes of these forces, of course, depend entirely on the contributing engine parameters and the particular gasket design used. Their magnitudes do, as mentioned previously, influence the system deflections in the assembly as well as the sealing stresses on the gasket. If the forces were known and an ideally rigid assembly (except for the bolts and gasket) were considered, the resulting deflection common to the bolts and gasket could be calculated as well as the change in the sealing stresses. The real case, however, usually presents the most challenge, so further discussion will be given that.

As mentioned earlier, the cylinder head gasket must fulfill the requirements presented under the basic theory of sealing and maintain the seal during combustion within the chamber. Recognizing that the sealing stress (on the gasket) is at a minimum during peak firing pressure, one might intuitively feel that the gasket should seal as long as any amount of sealing stress remains. Unfortunately, such is not indicated in laboratory work conducted at Victor. Rather, data observed agree very closely with the theory of static friction: $f = \mu N$.* That is, leakage past the gasket occurs when the sealed fluid pressure produces a force, F_H, against the edge of the gasket equal to or greater than the frictional forces, f, on the gasket opposing it. (FIG. 3). One would theorize that micromotion occurs between the gasket and mating surfaces at such a time which disturbs the plastically formed gasket surface, thereby allowing leakage to progress along

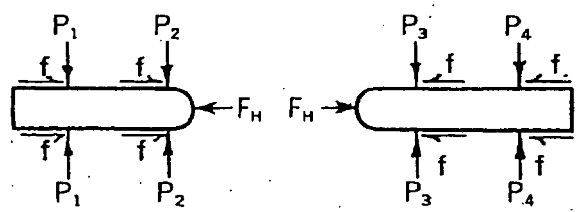

FIG. 3

various leakage paths. One can validly conclude, therefore, that a seal can be maintained provided the sealing load is always sufficient to allow a frictional component at least equal to the fluid force exerted on the edge of the gasket.

The apparent simplicity of the above is not intended to be misleading. As stated earlier, the sealing stress on the gasket is at a minimum during peak firing pressure. Therefore, the effective sealing load at this time must be sufficient to allow a frictional component adequate to resist the fluid force on the gasket edge. In order to achieve this, the rigidity of the cylinder head and block gasket surfaces must be considered as well as the initial load on the gasket, its thickness and its spring rate, the spring rate of the bolts, and the combustion force acting on the cylinder head. Precisely determining the effective values for each of these factors would be an almost insurmountable task, especially in view of the fact that each will assume a number of values depending on

*In the notation of this discussion, $f = \mu N$ would read $f = \mu \sum P_G$