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Rod Loading of Reciprocating Compressors

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ROD LOADING OF RECIPROCATING COMPRESSORS

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1. GENERAL

Rod loading is a major consideration for the designers and operators of reciprocating compressors such as the one illustrated in the cross-section. One has to ascertain that the net rod load does not exceed the design limitations placed upon the machine, which for the Superior compressors are shown in Table 1. These limitations are determined by consideration of many stresses: the rod compressive and tensile stresses, the rod column stresses, crosshead compressive and tensile stresses, the rod and crosshead thread stresses, and the crosshead pin (wrist pin) shear and bearing (bushing) stresses. These stresses are evaluated for their strength of material integrity including the stress at crosshead pin bushing. The crosshead pin bushing stress must be considered not only for its magnitude, but also, and more importantly, for its direction of application. Since the crosshead pin bushing stress (or load) is many times the limiting parameter in reciprocating compressor applications, we should take a closer look at its development, its effects, and its control.

2. CROSSHEAD PIN BUSHING LOAD (ROD LOAD) DEVELOPMENT

The load which is applied to the crosshead pin bushing is developed from two sources: the forces of inertia of the reciprocating piston, rod, and crosshead assembly, and the forces resulting from compression of the gas in the cylinder. Let us consider each of these forces individually, and then combine them to give us the total rod load.

a. Inertial Load

The inertial load is that force which develops as a result of the weight (mass) of the piston, rod, and crosshead assembly (including piston rings, nuts, crosshead pin, and balance weights) being in reciprocating motion. Figure 1 illustrates the inertial force developed by a 12.5" diameter 6" stroke cylinder with 250 pounds of reciprocating weight. This example also shows that the inertial load changes with the square of the compressor rotative speed. Figure 2 illustrates that the inertial load will also vary with the reciprocating weight. One should also notice that the inertial load is a "reversing" load in that it changes from tension to compression (and back to tension again) during one complete rotation of the crankshaft.

b. Gas Load

Figure 3 shows the gas load developed in a double acting 12.5" diameter 6" stroke cylinder operating from 100 psig suction pressure to 250 psig discharge pressure. The gas load on each end of the piston is determined by finding the pressure inside each end of the cylinder at various points in the stroke. This can be accomplished by approximating the compression process as an adiabatic process and using the relationship $P_2 = P_1\ (V_1/V_2)^k$. This pressure is then multiplied by the respective piston areas to give the head end and crank end gas loads. The total gas load is then found by adding the H.E. gas load and C.E. gas load. Figure 3 also indicates that the total gas load for a double acting cylinder contains a reversal.

c. Net Rod Load

(1) The net load applied to the crosshead pin bushing is found by the algebraic summation of the inertial load and the total gas load. Figure 4 illustrates the net rod load developed by the double acting 12.5" cylinder as well as the inertial and gas loads which comprise this rod load. Notice in Figure 4 that a load reversal of some 160° duration exists in every crankshaft revolution.

(2) Net rod loads for single acting cylinders are determined by the same procedure as for double acting cylinders. Figures 5 and 6 show the 12.5" cylinder in single acting head-end and single acting crank-end applications respectively. Both single acting applications have opened the inoperative end to suction pressure. As can be seen in Figures 5 and 6, single acting cylinders have load

reversals which are both shorter in duration and smaller in magnitude than in the majority of double acting cylinders. Notice that the reduction in reversal applies to both the total gas load as well as the net rod load. In fact, it can be seen that if the compression ratio were higher, a reversal for the gas load would not even occur, a very common phenomenon for single acting applications.

3. ROD LOAD EFFECTS

a. As mentioned previously, we must not only consider the magnitude of the rod load for design integrity, but we must also observe its direction of application (i.e. tension or compression). The direction of the rod load has a paramount effect on the lubrication of the crosshead pin bushing. When the load is being applied to one side of the bushing, some finite amount of clearance develops on the opposite side. This clearance, illustrated in Sketch 1, is filled with oil thereby lubricating and cooling that side of the bushing. In order to lubricate the other side of the bushing, a clearance must develop there also. A reversal in the direction of application of the load must occur for this to happen. And it is also evident that the magnitude and duration of this reversal must be such that a complete filling of this clearance space with oil can be effected. This is necessary to achieve adequate lubrication and cooling.

b. Reciprocating compressors operating with non-reversing loads are highly subject to bushing damage. Many past instances have shown that a bushing can fail within a very few minutes while operating under non-reversing loads. The failed bushing, such as the one shown in the Photograph No. 1 & 1A will exhibit severe wiping scars and scratches over approximately 120° of its I.D. and may very possibly show discoloration from overheating. This damage, which is characteristic of lubrication absence, will appear in the direction of the applied non-reversing load, or in the direction of the dominant load if the reversal is marginal.

c. The Superior crosshead pin bushing design, shown in Photograph No. 2, has incorporated helical oil grooves on the I.D. to aid the lubrication. While this design configuration reflects some performance improvements in non-reversing or marginal reversing load applications, it does not provide a suitable non-reversing load design that could be used for non-reversing condition. We must, therefore, continue to analyze each job to control the rod load in order that the non-reversing condition can be avoided. Let us look, then, at ways of controlling the rod load.

4. ROD LOAD CONTROL

a. The net rod load can be controlled by altering the constituents which comprise the rod load - namely the inertial load and the gas load. We certainly must evaluate how changes in each effect the rod load.

b. Inertial Load Changes - Examination of Figures 7 and 8 will reveal that by increasing the inertial force, the net rod load reversal also increases, both in magnitude and duration. By analyzing the inertial load formula shown in Figure 1, one can see that the inertial load can be increased without a major design change in the equipment by increasing either the rotative speed or the reciprocating weight. This gives us our first two means of controlling the rod load:

(1) Increasing the rotative speed of the compressor will increase the inertial load and the amount of the reversal. (See Figure 1)

(2) Adding weight to the crosshead and piston assembly will also increase the inertial load and the size of the rod load reversal. (See Figure 2).

Figures 7 and 8 show the effects of inertial load changes on the net rod load.

c. Gas Load Changes - The effect of the total gas load on the net rod load is quite different from the inertial effect. Increasing the total gas load may either increase or decrease the rod load reversal. In a single acting application, for instance, increasing the gas load on the operating end will decrease the reversal. This can be seen by reviewing Figures 5 and 6. On the other hand, however, increasing the gas load on the inoperative end of a single acting cylinder (i.e. opening the crank end of a small diameter cylinder to discharge pressure instead of suction pressure) would increase the reversal. Ample precautions must, therefore, be employed in making changes to the gas load. The changes we can make to the gas load are categorized into cylinder configuration changes and operating condition changes.

(1) Cylinder Configuration Changes

(a) The end selected for operation (in single acting applications) will effect the rod load reversal. Operating the crank end, which has a smaller piston area, will produce a smaller gas load and increase the reversal.

(b) Operating the cylinder in a double acting configuration will obviously increase the gas load and the reversal.

(c) The size of the cylinder bore will effect the reversal in either way. Smaller bores in double acting cylinders tend to decrease the reversal, but decreasing the bore in single acting configurations tends to increase the reversal.

(d) Decreased cylinder clearance will increase the volumetric efficiency and increase the gas load. In single acting situations, increased gas load will reduce the reversal; and in small diameter double acting cylinders, the increased gas load will expand the reversal.

(2) Operating Condition Changes

Reducing the compression ratio will decrease the gas load and usually improves the rod load reversal. This can be accomplished by either lowering discharge pressure or increasing the suction pressure. One must be careful in lowering the discharge pressure by itself, however, since this change can effect the reversal in either direction.

5. GUIDELINES

We are now fully aware of the necessity of avoiding non-reversing rod loads. And we have illustrated various ways of manipulating the loads to accomplish the needed reversal. We must now apply this knowledge to help us pinpoint non-reversing possibilities. Realizing the principles discussed above, we should be alerted for a non-reversing rod load when an application contains one of the following conditions:

a. Slow Speed Operation

Slow speed operation by itself is not necessarily a problem. But with other conditions present, slow speed could be a significant contributor to a non-reversing rod load.

b. Single Acting Operation

Non-reversing rod loads occur in single acting operation more than in any other situation. And single acting head end operation (SAHE) is always more susceptible to non-reversals than single acting crank end operation (SACE).

c. Small bore sizes in double acting cylinders approach a single acting condition and are non-reversal prone.

d. Low volumetric efficiencies (VE) often produce non-reversals. Low VE's result from high clearances, particularly in unloading sequences where clearance is deliberately added. When performing unloading, one should always remember that SAHE is more susceptible to non-reversals than SACE. The head end pockets should be opened first to avoid the non-reversal.

e. High compression ratios are apt to produce non-reversals.

f. High cylinder pressures are a natural for non-reversing rod loads. They usually mean high gas loads, small cylinder bores, and sometimes single acting operation - all of which are susceptible to non-reversals.

6. CALCULATIONS

Superior engineers have developed two Mark II Fortran programs which are used to evaluate rod loads. One program, which is primarily used to size compressor cylinders for specific applications, calculates the maximum compressive and tensile rod loads using the operating pressures external to the cylinder (i.e., external rod load). The external rod load is a close approximation of the actual rod load and indicates whether a reversal exists and if so its approximate magnitude and direction. By defining a limit for the external rod load, one can also determine if the magnitude of the rod load exceeds the design loading capability of the machine. This limit is properly set somewhat below the actual or internal rod load limit (i.e., rod load based on pressures internal to the cylinder) since pressure losses occur through the valves thereby producing a larger ratio inside the cylinder and hence a larger internal rod load. The use of an external rod load limit not only benefits the engineers at Cooper Energy Services to quickly evaluate rod loads, but it also enables customers to periodically examine their rod loads as operating conditions change.

The second program developed at Cooper Energy Services is used to calculate the internal rod load (both magnitude and direction) for any cylinder at any angle of crankshaft revolution from 0° through 360°. The program considers all of the necessary parameters which can effect the rod load including cylinder configuration (piston and rod diameters, stroke, rpm, reciprocating weight, clearances, valve losses, single or double acting, tandem and tail rod design, etc.), gas composition and compressibility, and all operating conditions (suction pressure and temperature, discharge pressure and temperature, barometric temperature, etc.). The product of this program is a print out, giving for each specified angle of crankshaft revolution, the HE and CE gas loads, the total gas load, the inertial load, and the net resultant of these components - the actual rod load. With this data readily available, the acceptability of the rod load's magnitude and reversal can easily be analyzed.

7. SUMMARY

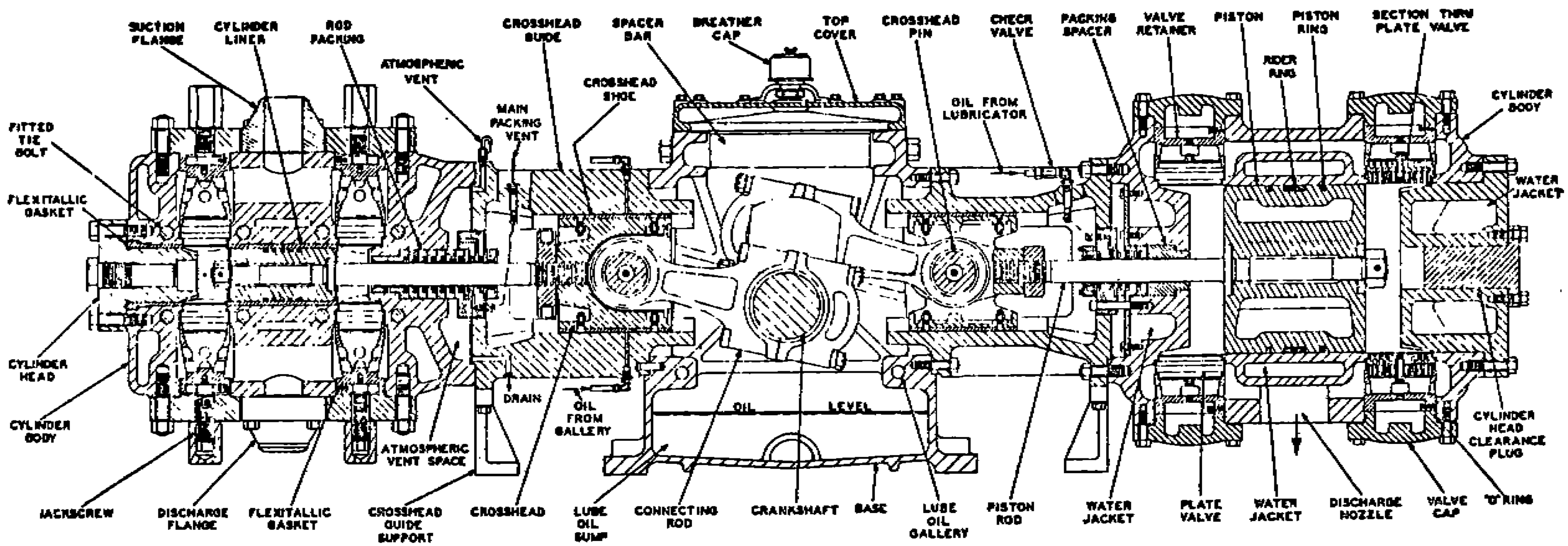
The purpose of this discussion was to provide a thorough understanding of rod loads and their effects. While gas loads and external rod loads can be used as a general guide, they alone do not completely define the actual rod load. Each job must be analyzed to control the rod load within the machine capabilities such that a satisfactory reversal is achieved. The tools provided here will help you to recognize the non-reversing possibilities and how to avoid their occurrence.

TABLE I

SUPERIOR COMPRESSORS

ROD LOAD LIMITS

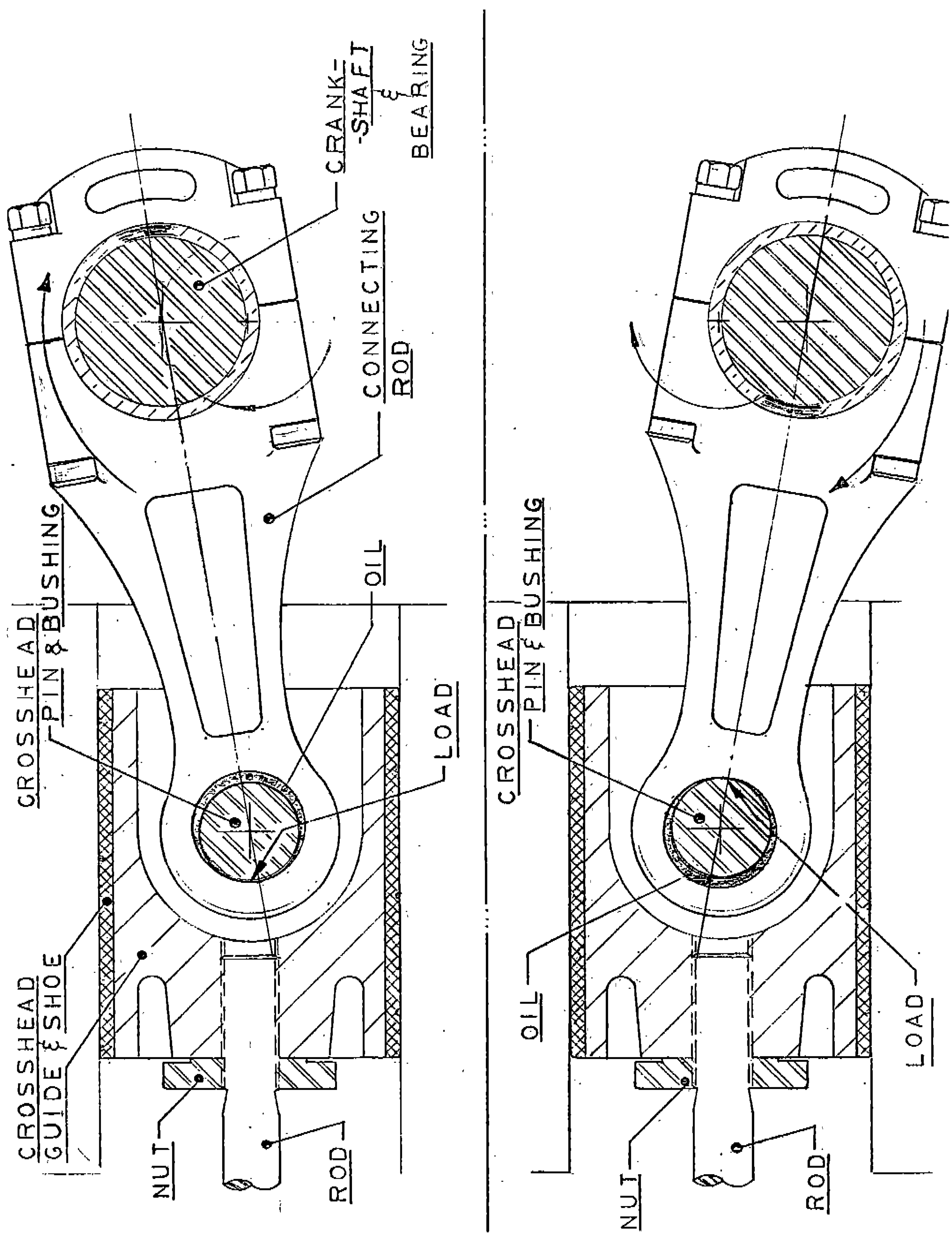
COMPRESSOR MODEL	INTERNAL ROD LOAD (LBS.) DOUBLE ACTING	INTERNAL ROD LOAD (LBS.) SINGLE ACTING	EXTERNAL ROD LOAD (LBS.) DOUBLE ACTING	EXTERNAL ROD LOAD (LBS.) SINGLE ACTING
W6	35,000	35,000	30,000	30,000
MW6	40,000	40,000	35,000	35,000



TYPICAL
HIGH PRESSURE CYLINDER

TYPICAL
LOW PRESSURE CYLINDER

TRANSVERSE CROSS SECTION
SUPERIOR COMPRESSOR


CRANK-SHAFT & BEARING
CONNECTING ROD
CROSSHEAD PIN & BUSHING
OIL
LOAD
CROSSHEAD GUIDE & SHOE
NUT
ROD
CRANK-SHAFT & BEARING
CROSSHEAD PIN & BUSHING
OIL
LOAD
NUT
ROD

SKETCH 1

INERTIAL LOADS

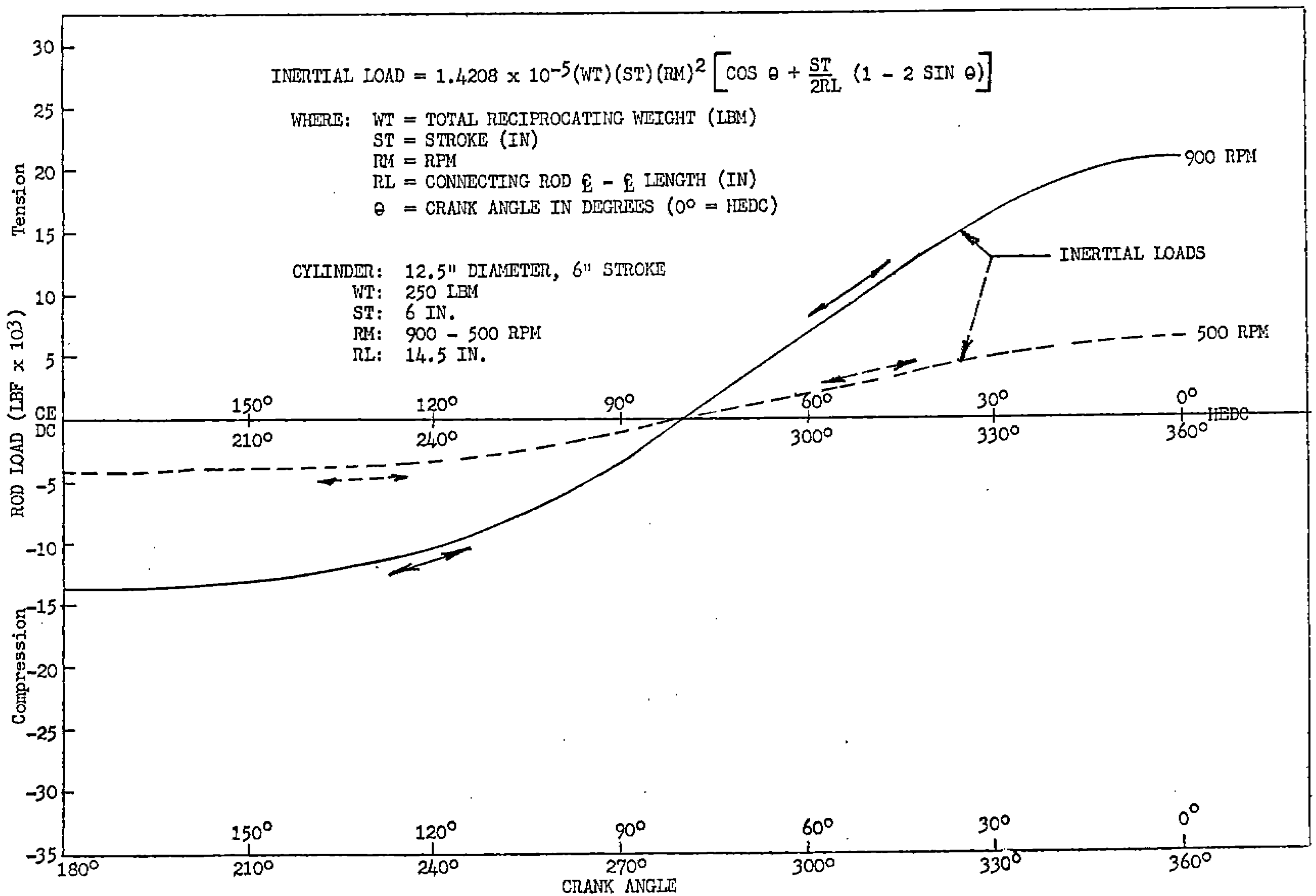


FIGURE 1

INERTIAL LOADS

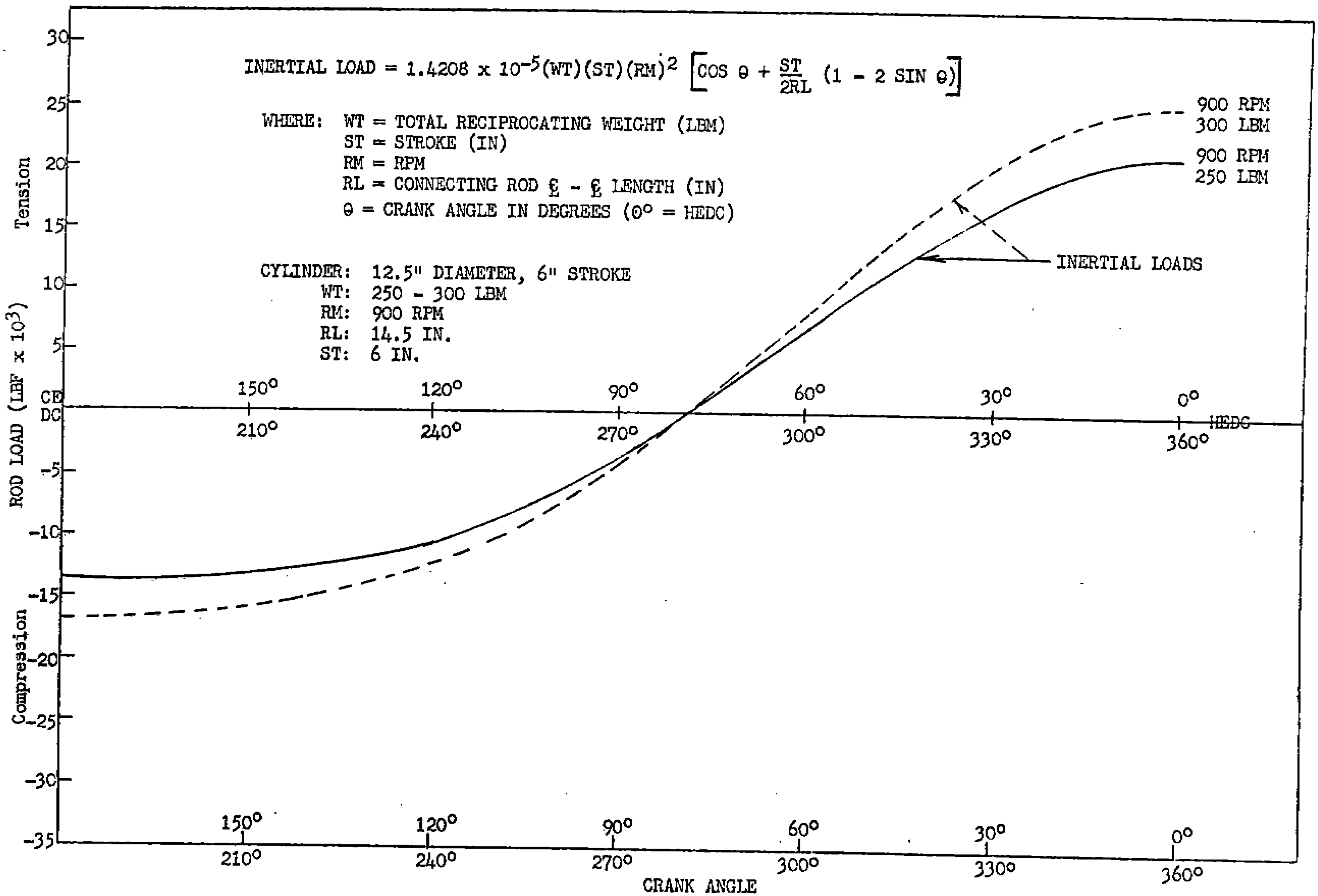


FIGURE 2

DOUBLE ACTING GAS LOADS

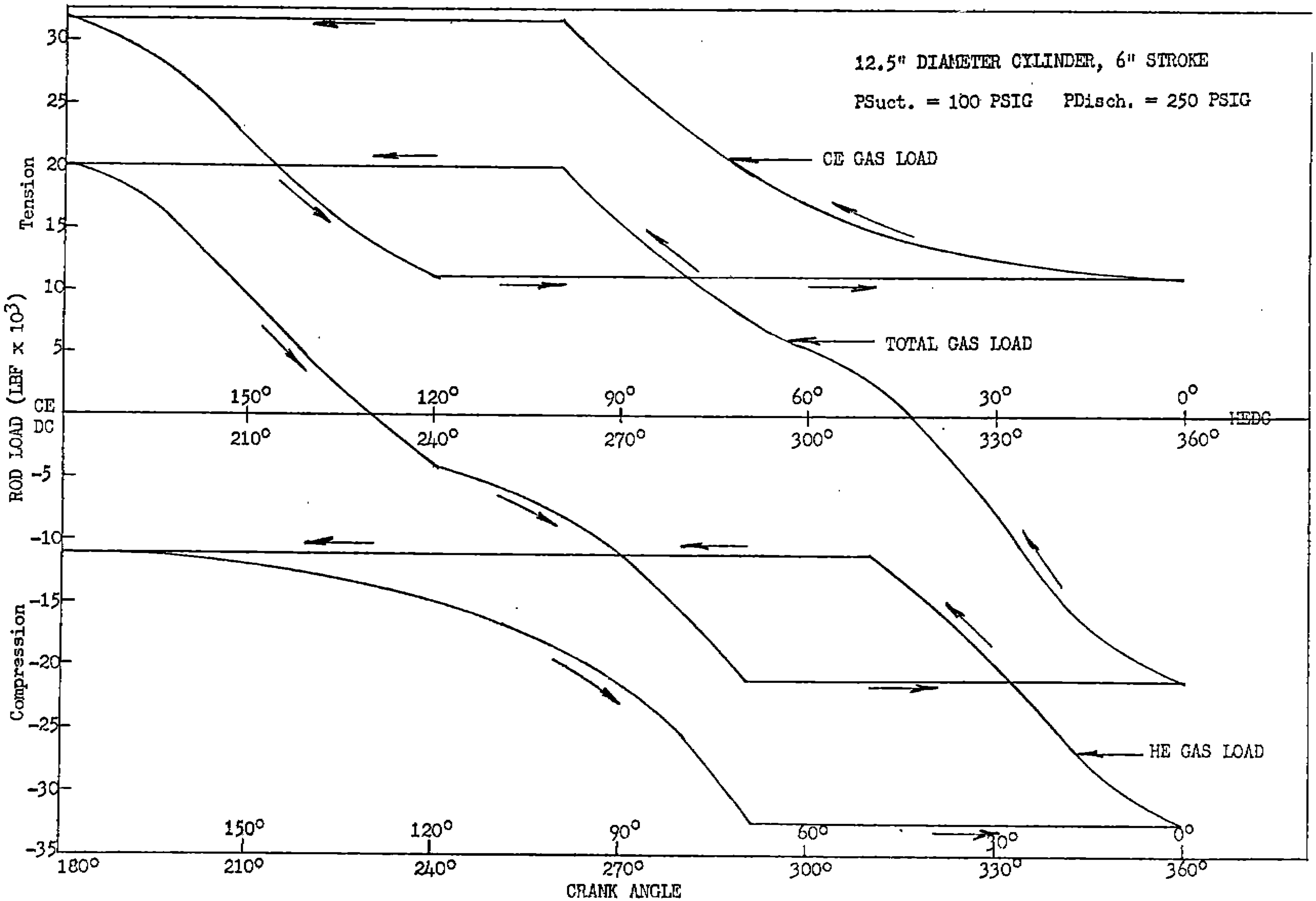
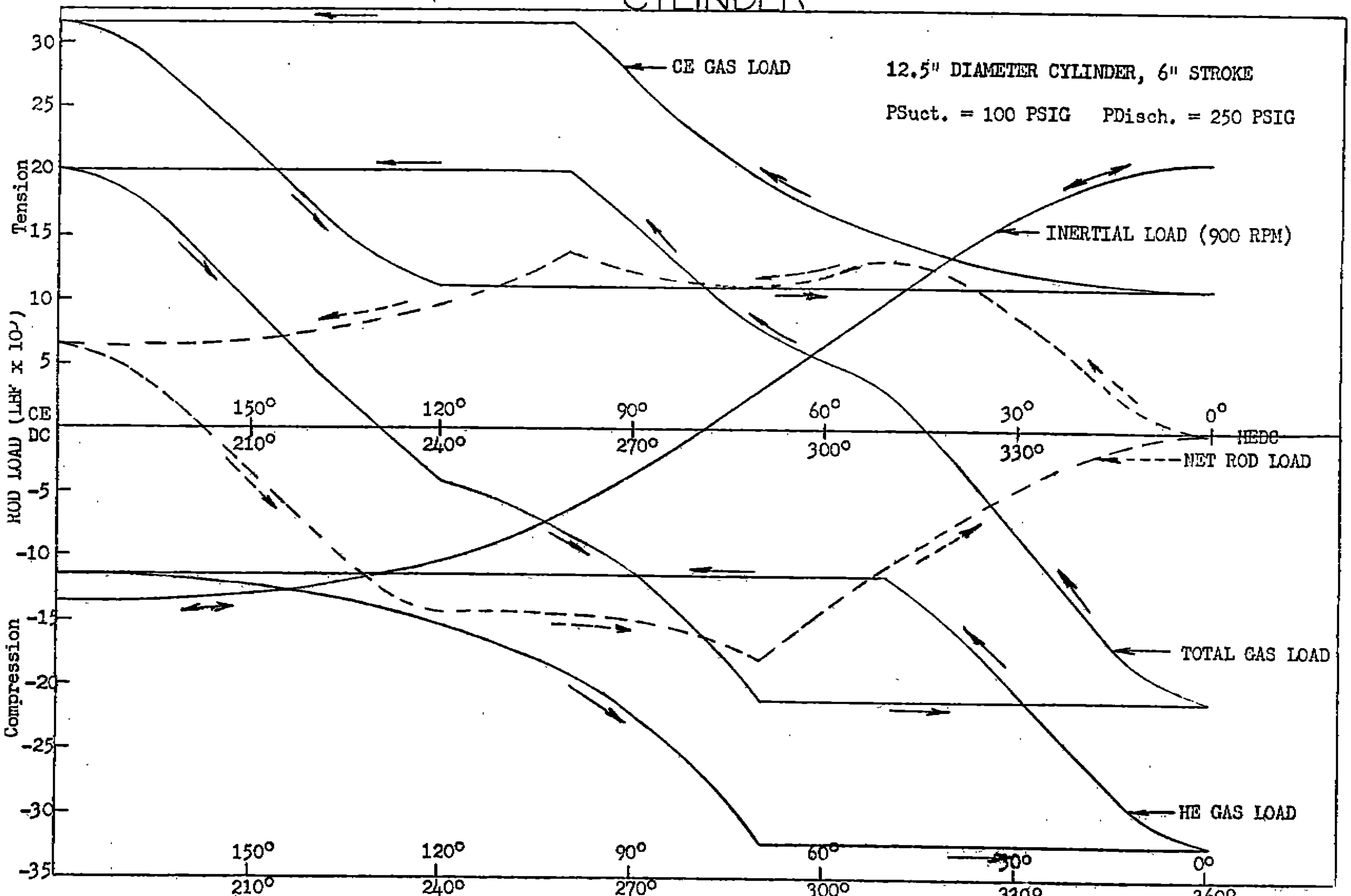


FIGURE 3

ROD LOADING DOUBLE ACTING CYLINDER



CRANK ANGLE

FIGURE 4

ROD LOADING SINGLE ACTING CYLINDER
CRANK END OPEN TO SUCTION PRESSURE

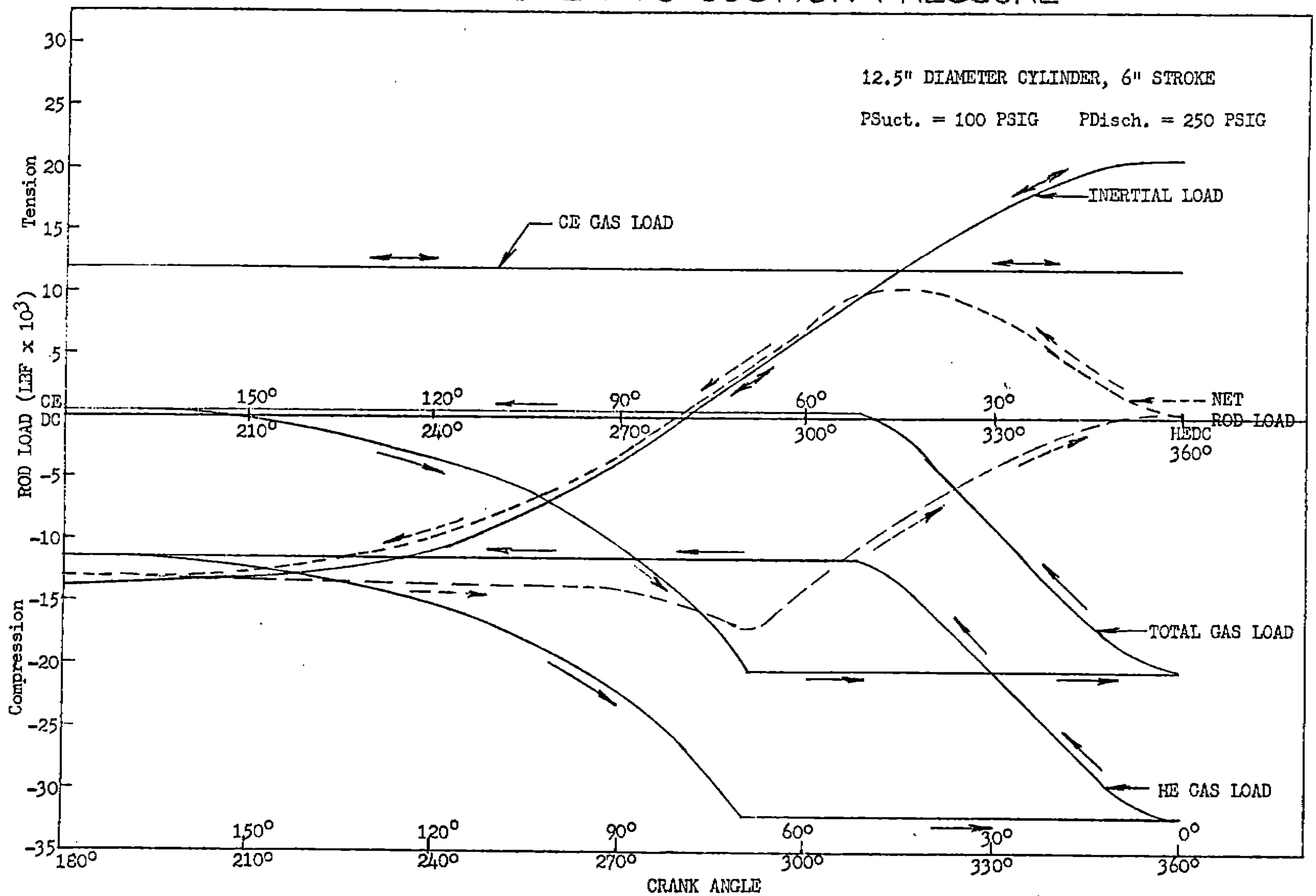


FIGURE 5

ROD LOADING SINGLE ACTING CYLINDER
HEAD END OPEN TO SUCTION PRESSURE

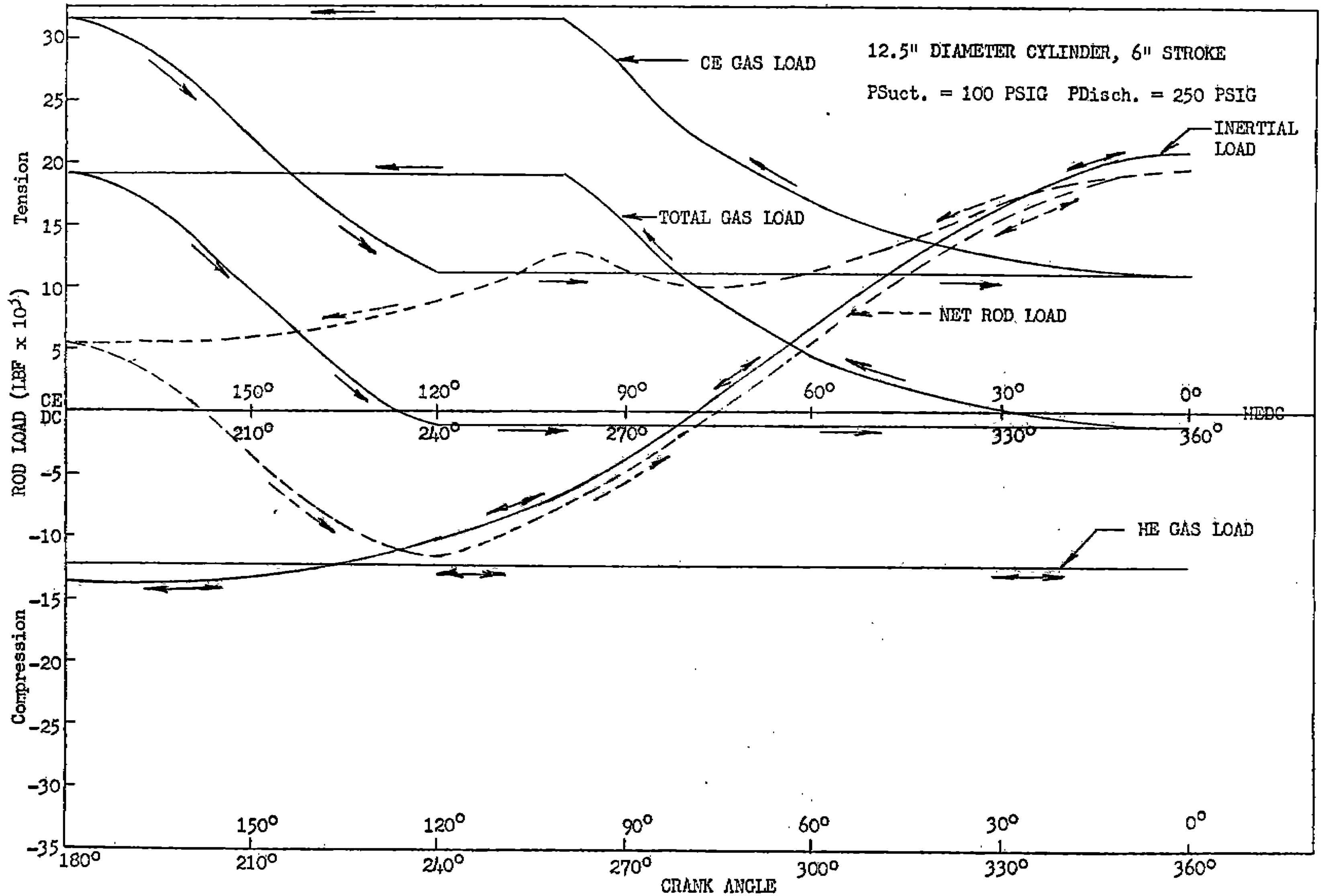
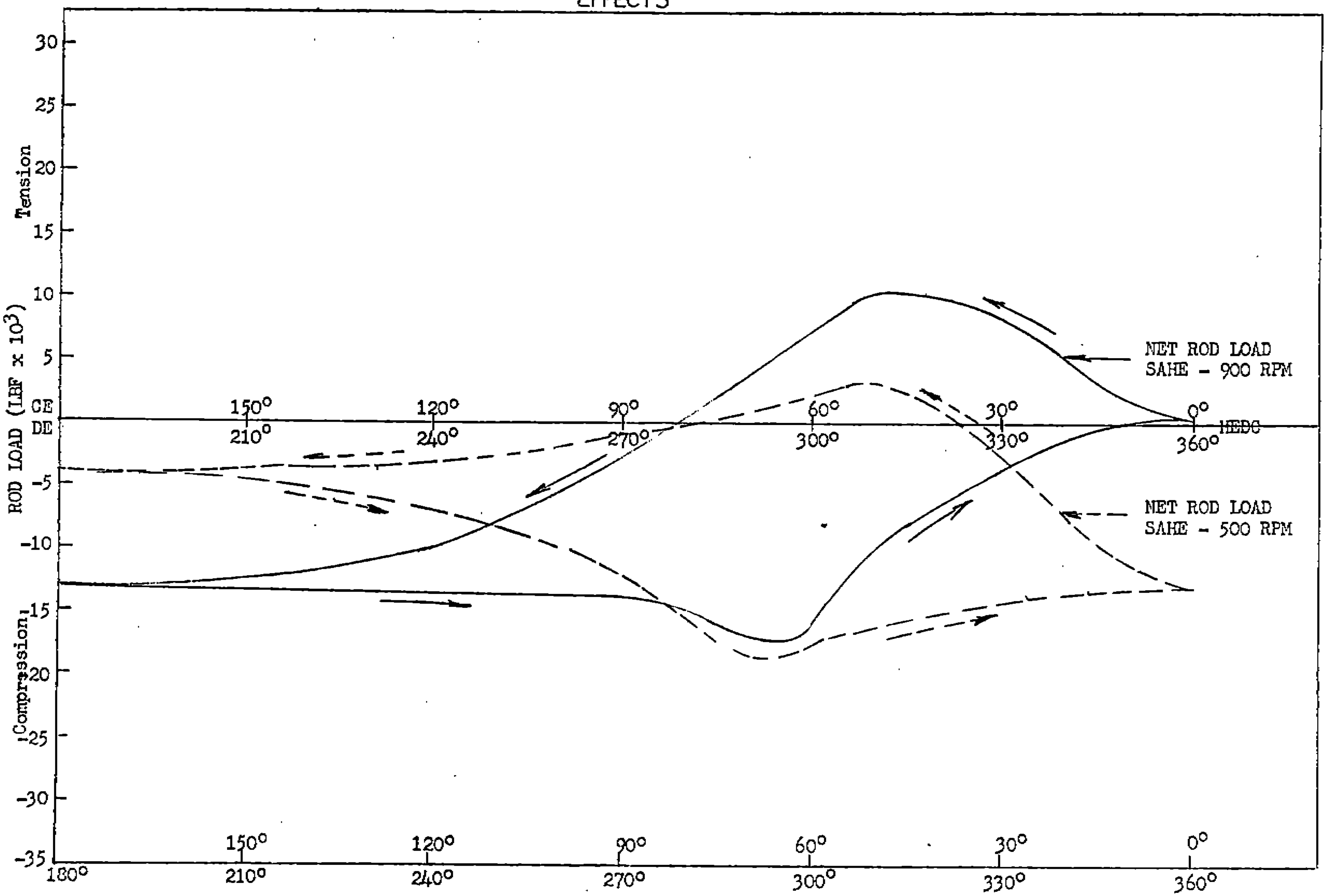


FIGURE 6

NET ROD LOADS SINGLE ACTING CYLINDER CRANK END OPEN TO SUCTION PRESS. INTERNAL CHANGE EFFECTS



CRANK ANGLE

FIGURE 7

NET ROD LOADS SINGLE ACTING CYLINDER
HEAD END OPEN TO SUCTION PRESS. INTERNAL CHANGE
EFFECTS

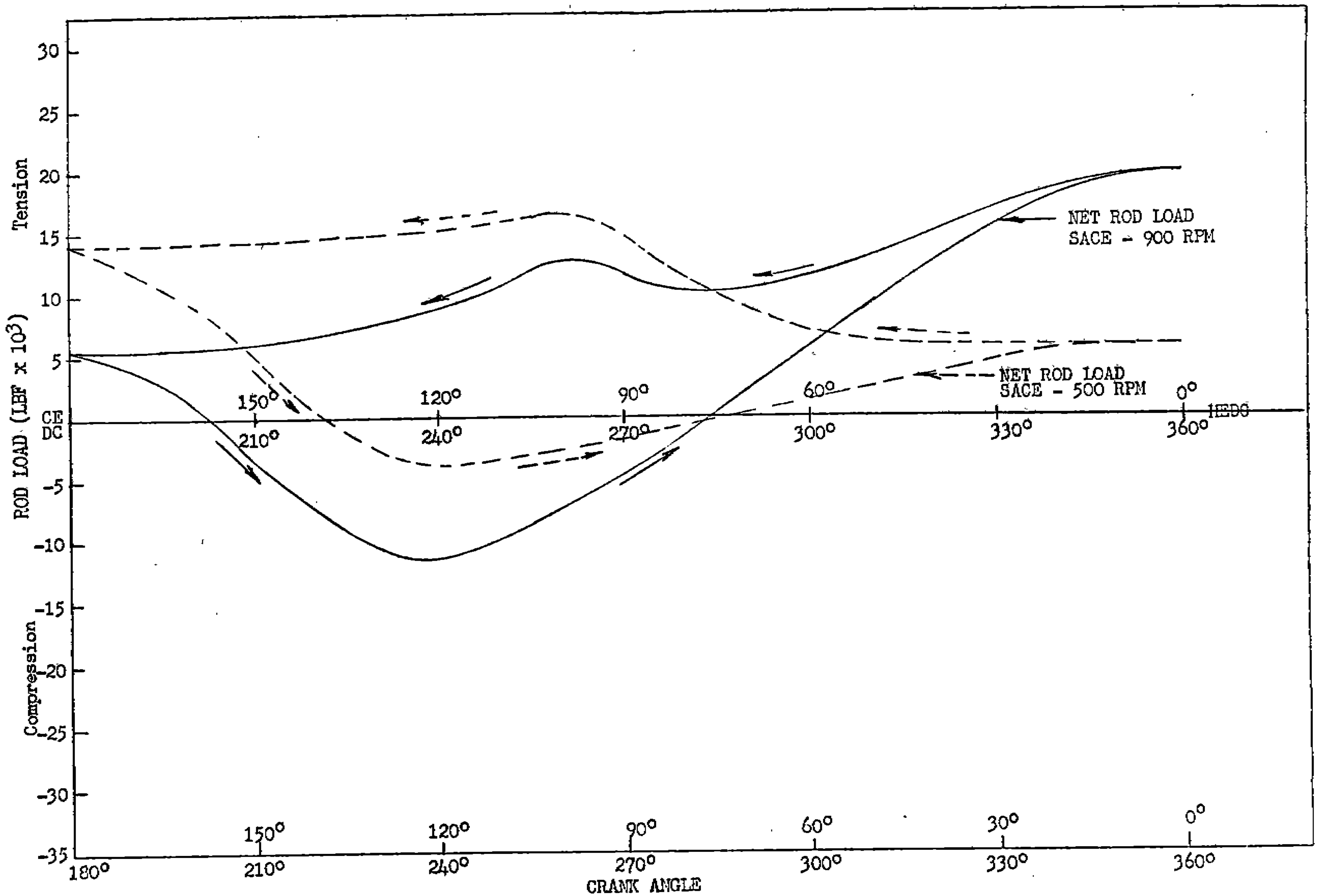
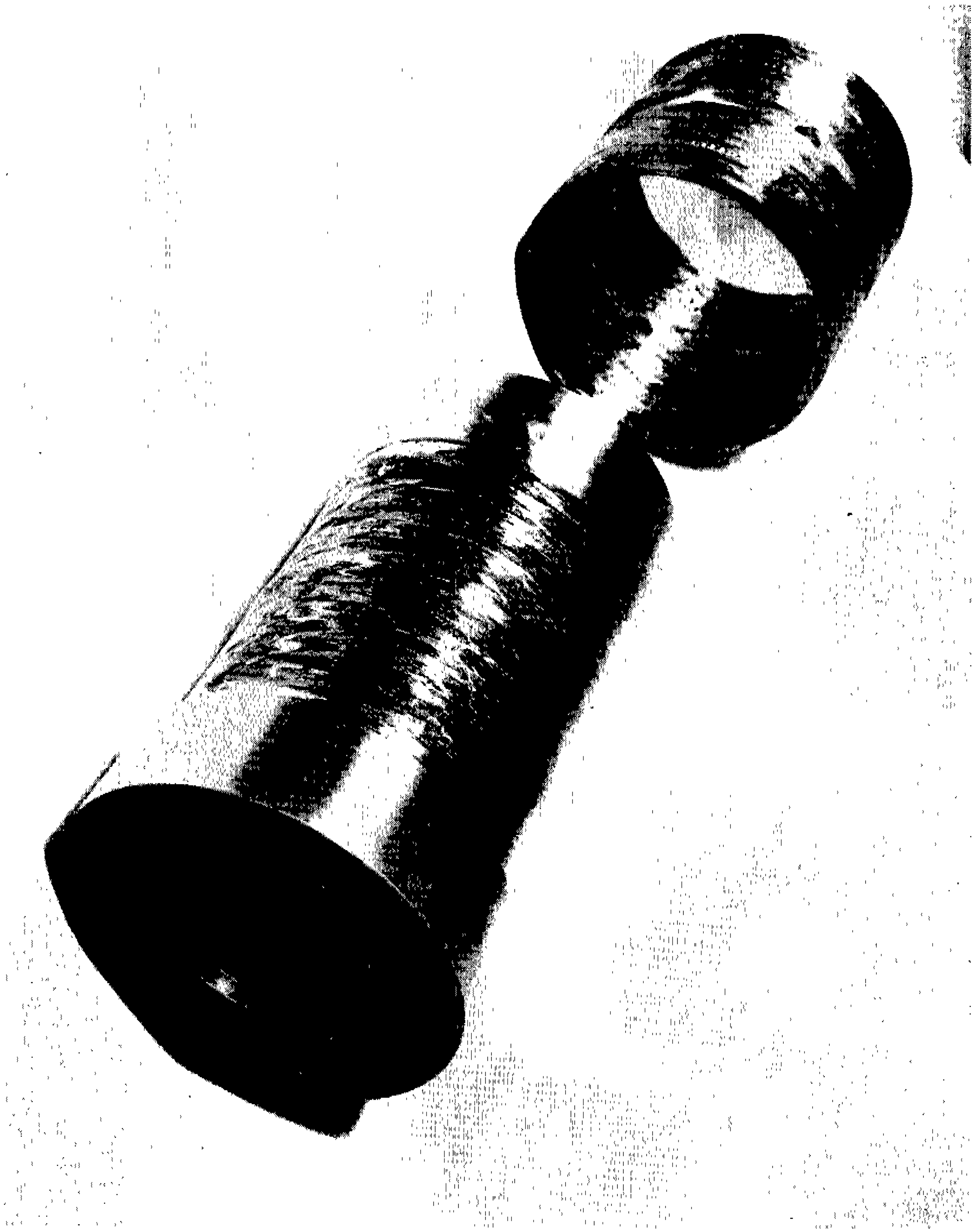
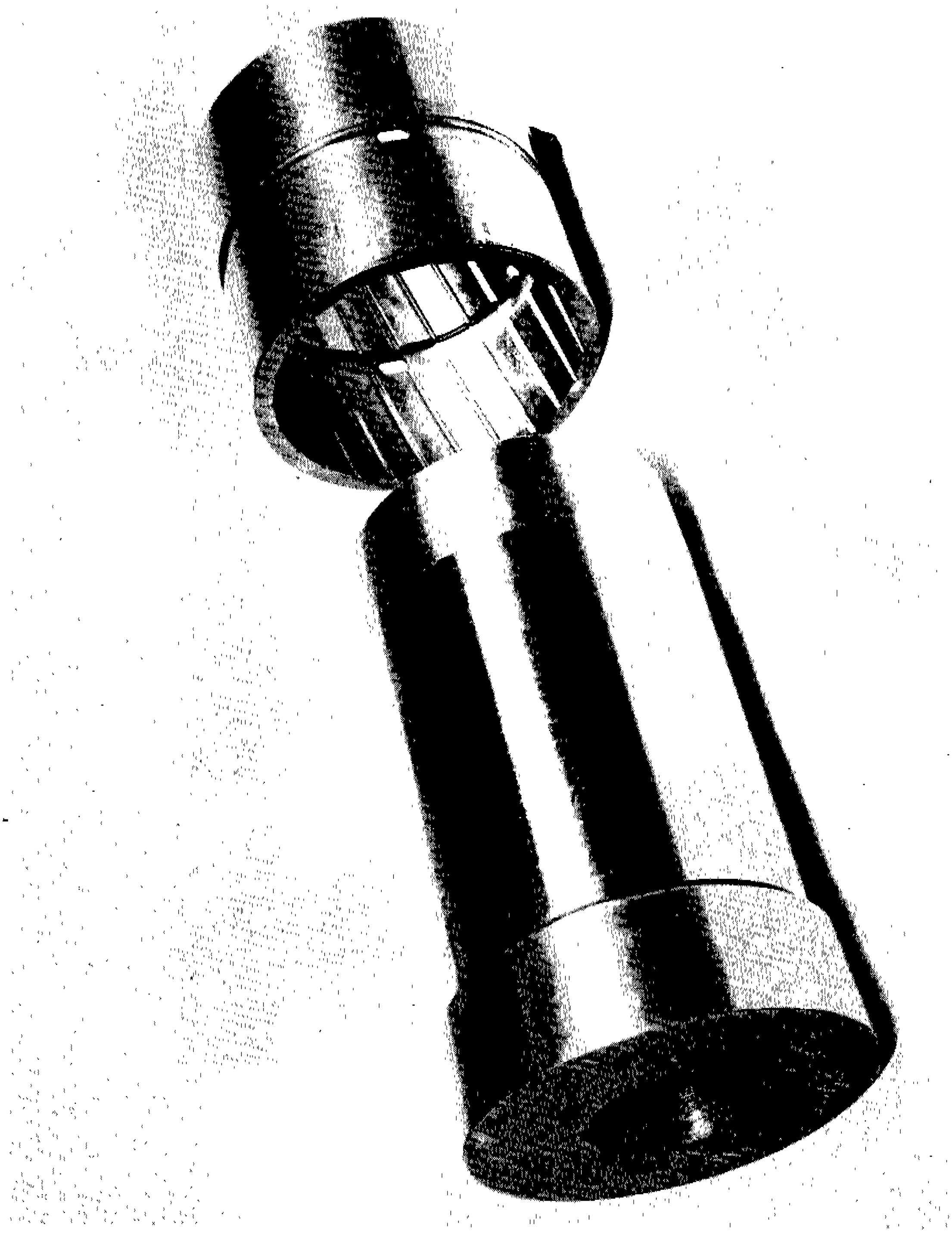


FIGURE 8



PHOTOGRAPH NO. 1

PHOTOGRAPH NO. 1A



PHOTOGRAPH NO. 2