Analytic Prediction of the Startability of Reciprocating Refrigeration Compressors Including Comparison with Experiment

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ABSTRACT

Crankshaft speed is predicted as a function of time, geometry, inertial characteristics, operating conditions and initial position for a reciprocating compressor during start-up. The governing equations are obtained by applying Newton's second law to the various components of the slider-crank mechanism. The results of the analysis were verified by experimental observation.

INTRODUCTION

The ability of a reciprocating refrigeration compressor to start under conditions likely to be encountered during field service is one of its most important performance prerequisits. A successful start requires but a fraction of a second. Only a few electrical cycles are needed in order to accelerate the crankshaft from rest to velocities within 5 percent of the synchronous speed of the motor. A failure to start requires even less time as the crankshaft rotates only a fraction of a revolution before coming to rest.

The occurrence of a start or a stall is dependent on a number of parameters including suction and discharge pressures, flow losses, motor torque and compressor geometry. The initial position of the crankshaft-connecting rod-piston mechanism is an equally important consideration. These parameters and the relationships that exist between them as expressed by the appropriate kinematic relationships and Newton's second law of motion may be used to produce a reliable prediction of startability.

The cyclical torque requirements of reciprocating compressors are not uniform. The reexpansion stroke produces a negative torque demand on the motor while the latter stages of the compression stroke generate a large positive torque requirement. The torque output of the motor is essentially a function of crankshaft speed rather than cyclic position. In general, a mismatch exists between the instantaneous torque supplied by the motor and that required by the compressor. The torque supplied in excess of the torque required results in an acceleration of the crankshaft; i.e., an increase in the kinetic energy stored in the mechanism. The excess torque accumulated in this fashion is available for utilization at a later time should it be needed. A stall occurs when the instantaneous torque supplied and the reserve of past excess torque in the form of kinetic energy are less than the present torque demand. The initial position of the crankshaft determines the amount of kinetic energy that can be stored in the mechanism prior to the point in the cycle where the torque required exceeds that developed by the motor at a given instant. It is a very important consideration, it is the principal reason why both "starts" and "stalls" occur under identical pressure conditions.

MATHEMATICAL MODEL

The analysis which follows is based on numerous simplifying assumptions that allow the mathematics to be reduced to workable proportions. Several of the more important idealizations are listed below:

1) A start or stall occurs sufficiently fast that the pressure conditions observed in the suction and discharge lines prior to the time the motor is energized remain essentially constant.

2) The continuous physical processes occurring within a reciprocating compressor can be accurately modeled by a sequence of discrete small incremental steps taken one at a time.

3) The idealized pressure-volume diagram yields a sufficiently accurate representation of refrigerant properties.

4) Flow losses can be adequately represented

by increasing the apparent discharge pressure and reducing the apparent suction pressure by some small finite constant amount.

The resultant procedure has been fine tuned to the point where a Fortran program embodying this sequence of calculations requires approximately 15 min. of PDP 11/50 time to map out cyclic pressure conditions defining the stall region for a given motor and compressor combination.

The slider crank mechanism consists of a piston, connecting rod and crankshaft, as shown in Figure 1 for a single cylinder unit.

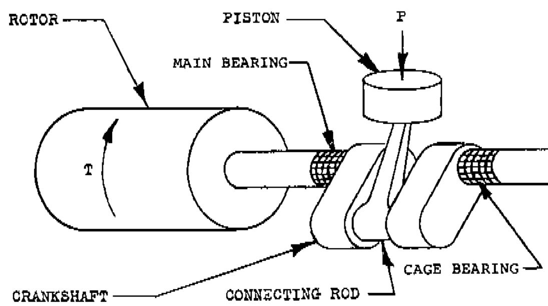

Figure 1

A detailed analysis of the cyclic time dependent nature of this mechanism may be found in Reference 1. Equations (1), (3), (4), (5) and (6) from this source constitute a set of five simultaneous linear equations in terms of five unknowns detailing the magnitude and direction of the forces that act between the piston and connecting rod, the connecting rod and crankshaft, and the instantaneous acceleration of the crankshaft. These equations are reproduced below:

$$F_{3PX} - M_pA_p - P + S_{WP}M_{WP}F_{3PY} - V_pC_{WP} = 0 \quad (1)$$

$$F_{23X} - F_{3PX} - M_{cr}A_{cgx} = 0 \quad (3)$$

$$F_{23Y} + F_{3PY} - M_{cr}A_{cgy} = 0 \quad (4)$$

$$F_{23X}\,(M \sin\phi - S_{23X}M_{23}R_{23})$$

$$-F_{23Y}(M \cos\phi + S_{23Y}M_{23}R_{23})$$

$$+F_{3PX}[(L-M)\sin\phi + S_{3PX}M_{3P}R_{3P}]+(R_{3P}w_3C_{3P}) \quad (5)$$

$$+F_{3PY}[(L-M)\cos\phi + S_{3PY}M_{3P}R_{3P}]-[R_{23}C_{23}(w_2-w_3)]$$

$$+ I_{cr}A_3 = 0$$

$$F_{23X}[R \sin\theta - S_{23X}(M_{23}R_{23} + M_{CSB}R_{CS})]$$

$$-F_{23Y}[R \cos\theta + S_{23Y}(M_{23}R_{23}+ M_{CSB}R_{CS})] \quad (6)$$

$$-I_{csr}A_2 + T_m - 2R_{CSB}w_2C_{CSB} - R_{23}C_{23}(w_2-w_3)=0$$

This analysis can be extended to handle multiple cylinder units by realizing that equations (1), (3), (4), and (5) are repeated for each piston and connecting rod pair while equation (6) is expanded to include F23X and F23Y terms for each cylinder. Thus, a compressor with N cylinders can be modeled by 4N + 1 equations.

The torque supplied by an induction motor may be calculated as a function of the impressed voltage and the fundamental constants describing the motor, as described in Reference 2. Alternately, measured performance data can be utilized as outlined in Reference 3 to provide a means of calculating motor output as a function of speed and voltage.

CALCULATION PROCEDURE

The calculation procedure is begun by assuming an initial set of conditions and tracking the motion of the mechanism through time until rotation ceases or the crankshaft speed exceeds the break-away speed of the motor.

If the first sequence of calculations results in a run, the discharge pressure is increased an arbitrary amount ΔP; and the computations are repeated until a stall occurs. (The procedure is just the reverse if the first calculation is a stall). Thus, for a given start position, two discharge pressures differing in absolute value by ΔP are found such that within this range lies the dividing line between a start and a stall. The computations are repeated for a new "starting" position $\Delta\theta$ away from the last until all possible starting positions have been evaluated. A curve illustrating the results of this analysis for a one-cylinder compressor is shown in Figure 2. Figure 3 depicts similar data for a two-cylinder unit.

EXPERIMENTAL VERIFICATION

Laboratory verification of Figure 2 was accomplished by observing the initial crankshaft position of a test unit prior to energizing the motor. Different values of discharge pressure were set before applying power to the motor. In this way, it was possible to determine the dividing line between pressure conditions producing a start and stall.

The double cross-hatched, U-shaped region in Figure 1 was determined experimentally. It is the observed transition region between starts and stalls. The solid dark line at the bottom of this area is the calculated curve. The agreement between the theoretical prediction and that observed in the laboratory is more than satisfactory.

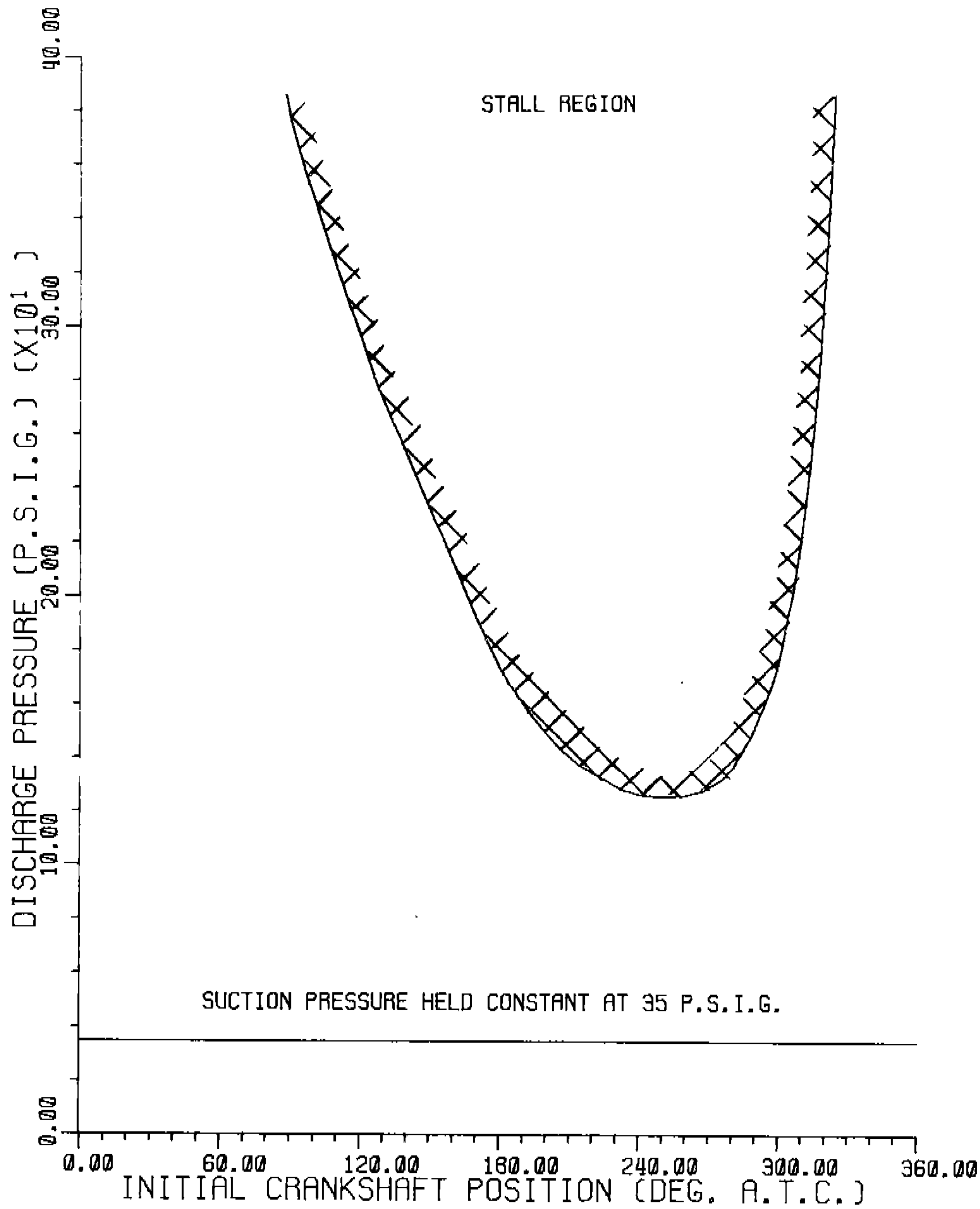

Figure 2

It was also discovered that the position at which the crankshaft came to rest upon termination of a successful start was not randomly distributed throughout the cycle. The stopping position was found to vary from one test to the next. Values were observed to range between 160° and 220° on the X scale of Figure 2. The most frequently observed stopping point was 195°. This raises the possibility of a "chicken and egg" type argument with regard to starts and stops.

CONCLUDING REMARKS

The ability of a motor-compressor combination to start under a pressure-imposed load is dependent on the initial position of the power mechanism. The relative importance of this phenomenon is diminshed as the number of phased cylinders increases, all else being equal. Startability is a "sure thing" for pressure conditions lying below the minimum point of the stall curve. More stringent loads impose a need for statistical measures and involve a probability of experiencing either a start or a stall.

The seemingly steady behavior of reciprocating machinery is a periodic transient phenomena, i.e., considerable change occurs within the period of a cycle although each complete cycle is the same as all others. The same techniques may be used to predict both inter and intra-cycle variations.

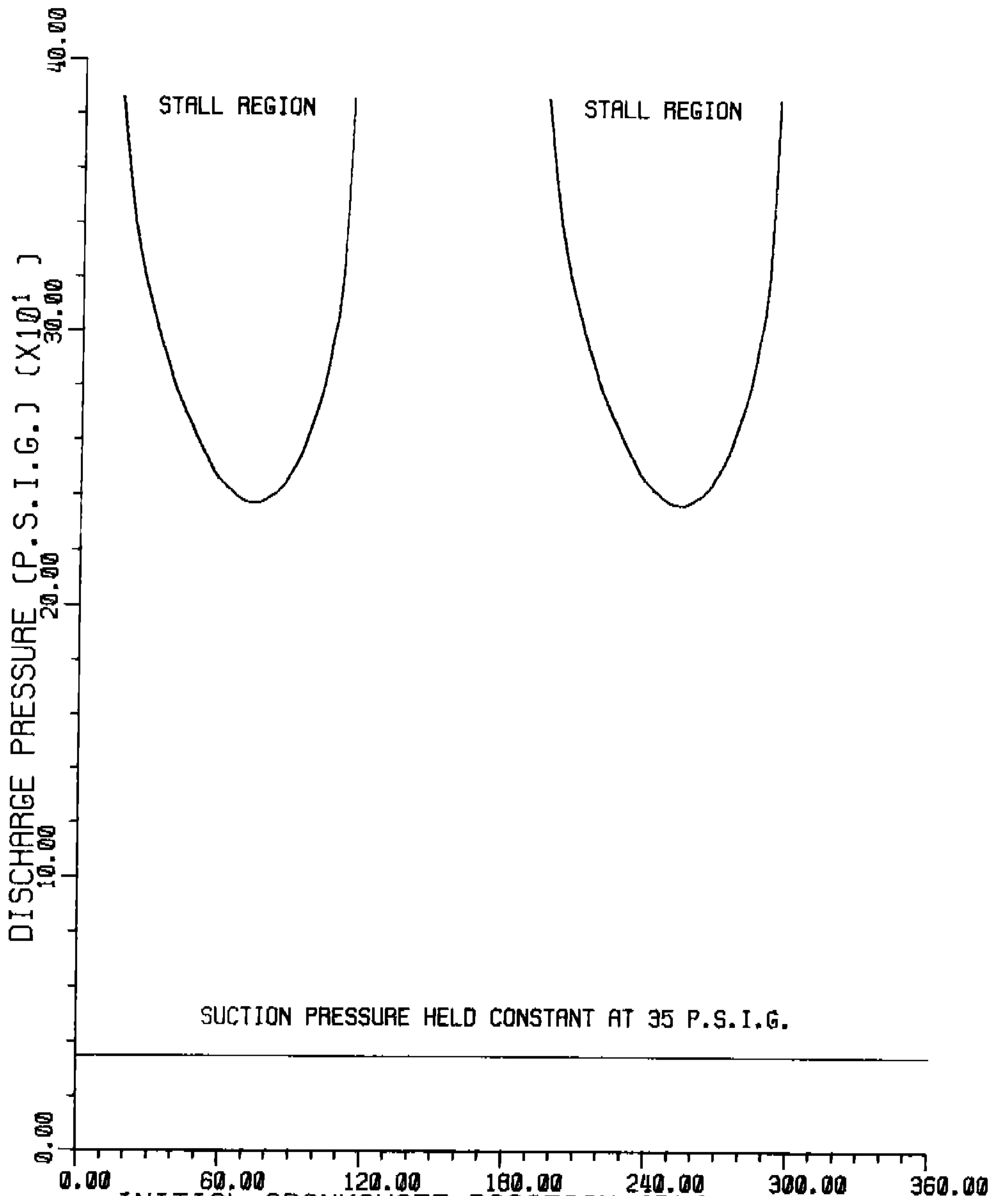

Figure 3

LIST OF SYMBOLS

A_{cgx} Acceleration of connecting rod center of gravity in the x-direction

A_{cgy} Acceleration of the connecting rod center of gravity in the y-direction

A_p Acceleration of the piston in the x-direction

C_{23} Coefficient of viscous friction in the bearing between the connecting rod and crankshaft.

C_{CSB} Coefficient of viscous friction in the crankshaft bearings

C_{WP} Coefficient of viscous friction in the bearing between the piston and the connecting rod

F_{23X} Force exerted on the connecting rod in the x-direction by the crankshaft

F_{23Y} Force exerted on the connecting rod in the y-direction by the crankshaft

F_{3PX} Force exerted on the piston in the x-direction by the connecting rod

F_{WP} Force exerted on the piston normal to the wall

I_{cr} Connecting rod moment of inertia about its center of gravity

I_{csr} Moment of inertia of the crankshaft and rotor about their axis of rotation

L Length between connecting rod bearing centers

M Distance from crankshaft bearing center to connecting rod center of gravity

M_{23} Coefficient of sliding friction in the bearing between connecting rod and crankshaft

M_{3P} Coefficient of sliding friction in the bearing between the piston and the connecting rod

M_{cr} Connecting rod mass

M_p Piston mass

M_{WP} Coefficient of sliding friction between the piston and the wall

P Force exerted on the piston due to gas pressure

R Crankshaft throw

R_{23} Radius of the bearing between the connecting rod and crankshaft

R_{3P} Radius of the bearing between the piston and connecting rod

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