

THE MECHANICAL DESIGN OF A VAPOR COMPRESSOR FOR A HEAT PUMP TO BE USED IN SPACE

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INTRODUCTION

A heat pump is presently being developed for use in Spacelab as a stand-alone refrigeration unit, as well as within a fluid loop system, that will provide an active thermal control for payloads. The sponsor of this work, the European Space Agency, has established the following specifications for the heat pump:

- Heat removal rates at the source (payload or fluid loop) from tens of watts at subzero (degrees C) temperatures to a few hundred watts during the initial cooling of payloads from room temperature or levels above it
- Heat source temperatures from room temperature down to -30°C , arbitrarily selectable
- Heat-sink fluid temperatures at condenser inlet between $+20^{\circ}\text{C}$ and $+40^{\circ}\text{C}$
- Minimum reasonable achievable power consumption of the heat pump for each set of the parameters: heat source temperature, heat removal rate at the source, and heat-sink fluid temperature at condenser inlet

Based on a comparison of different heat pumping schemes including gas cycles, vapor cycles, absorption cycles, and thermoelectric (Peltier) elements, it was decided to meet these requirements with a reversed Carnot cycle heat pump using Freon 12 as working fluid and incorporating a one-cylinder, reciprocating compressor.

For weight reasons and to avoid large deviations from a uniform rotation, the maximum crankshaft speed was fixed relatively high at 1000 rpm. The specified cooling rates then made it necessary to select a cylinder volume of 10 cm^3 , which was obtained with a bore of 40 mm and a stroke of 8 mm.

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The engineering model of the vapor compressor has been built and tested in a Freon 12 test loop [1] . Figs. 1 and 2, established for the two heat-sink fluid temperatures of 20°C and 40°C, respectively, show the expected performance of a heat pump incorporating our vapor compressor, based on its experimentally determined performance characteristics.

SOME SPECIAL ELEMENTS OF THE MECHANICAL DESIGN

A section through the vapor compressor is presented in Fig. 3 in which some of the special elements of the mechanical design are called out.

Means for Reducing the Nonuniformity of Rotation

A problem of single-cylinder reciprocating compressors is the nonuniformity of rotation of the crankshaft. In the case of a vapor compressor of a heat pump using Freon 12, this problem is accentuated by the fact that in the range of heat source temperatures of interest from -30°C to +20°C, the saturation pressure - which is close to the minimum pressure in the cylinder - varies from about 1 bar to 5.8 bar. Therefore, even if the compressor were operating in a 1-bar atmosphere, the pressure in the cylinder would generally be well above the ambient pressure during the downward motion of the piston following a compression phase, and the torque in the crankshaft would thus vary from positive values to relatively large negative values during a revolution. To reduce the torque variation, we have incorporated a compression spring between piston and crankshaft housing. Energy is stored in the spring through the latter's compression during the downward motion of the piston, and this energy is returned to the piston during the upward motion. Thus, the spring's effect is similar to that of a flywheel.

Lubrication Under Null Gravity Conditions, Separation of Refrigerant from Lubricants, and Associated Problems

A number of design features have been introduced because of problems associated with the operation in a zero-g environment. The absence of gravity makes oil lubrication impractical because no defined oil level can be maintained. Hence, grease lubrication has been selected in ball bearings and needle bearings. Dry lubrication with a loaded PTFE is employed in some areas, such as the guidance of the tappet and of the stem of the discharge valve and for the piston ring. No lubrication at all is used between tappet and camshaft, whereby the gliding contact between these elements has been replaced by a practically pure rolling contact with the addition of a roller to the tappet head.

Since lubricants exposed to the refrigerant Freon 12 would absorb it at a significant rate [2] (whereby their performance would be impaired if they were not purified from time to time as in fact greases could not be), it was

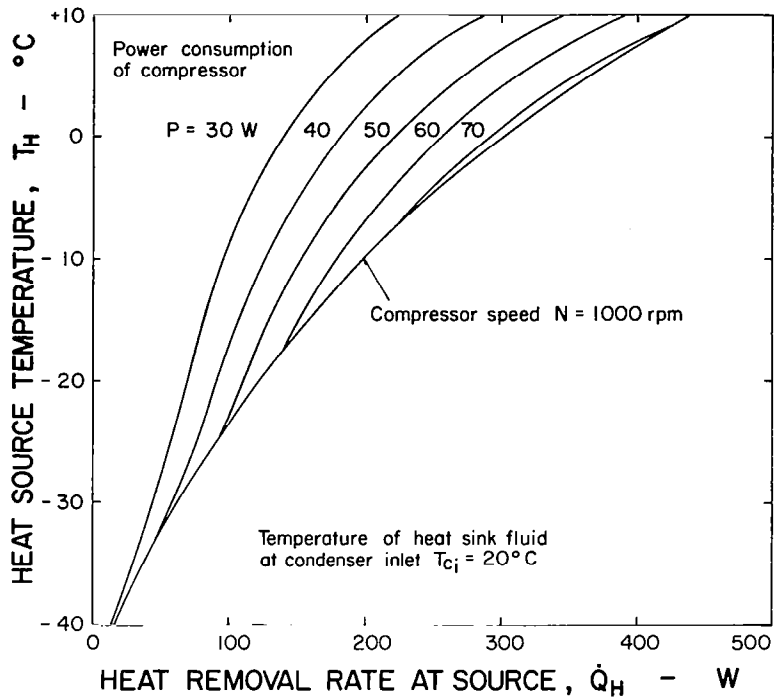


Fig. 1. Performance map of heat pump based on experimentally determined compressor characteristics ($T_{ci} = 20^\circ\text{C}$).

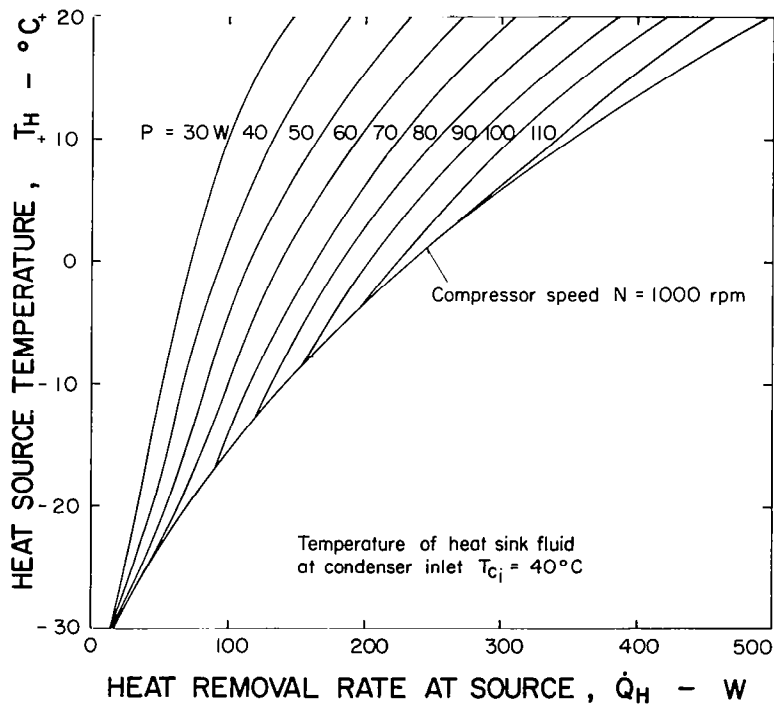


Fig. 2. Performance map of heat pump based on experimentally determined compressor characteristics ($T_{ci} = 40^\circ\text{C}$).

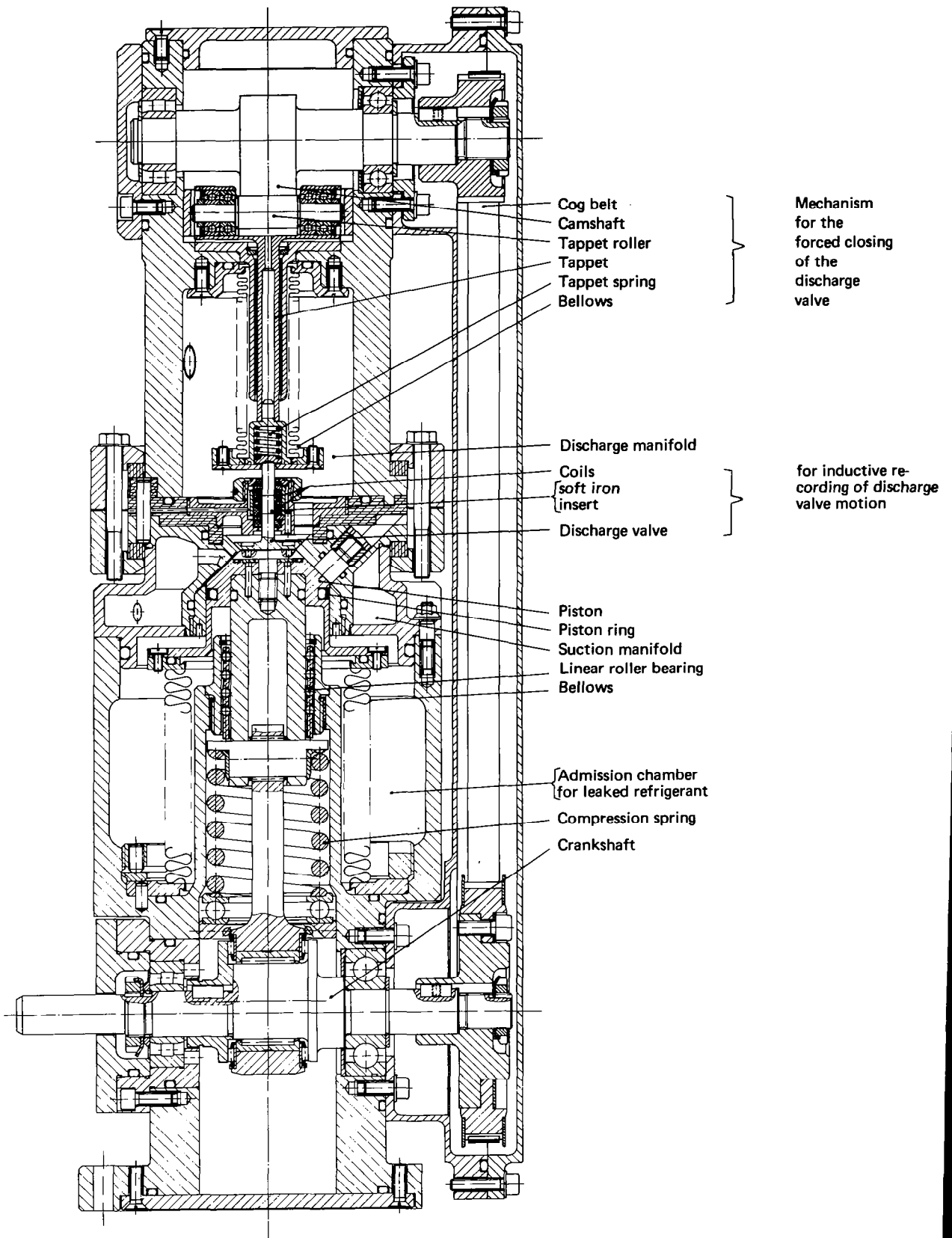


Fig. 3. Section through vapor compressor.

decided to seal off completely the portion of the Freon loop within the compressor from its immediate surroundings. To achieve a hermetic **containment** of the refrigerant, we have introduced metal bellows between the piston and the lower portion of the compressor housing and between the tappet and the discharge manifold. Both bellows have been subjected to extensive fatigue tests which they passed successfully [3] .

The use of a dry-lubricated piston ring made it necessary to reduce the clearance between piston and cylinder surface below what is standard practice in compressors with oil-lubricated piston rings. Hence a strictly axial piston movement, without any skewing, must be achieved in our compressor. This requirement led to the incorporation of a grease-lubricated, **linear roller bearing**. The rollers of this bearing are contoured such that they match the cylindrical surfaces of piston shaft and guiding bush.

THE SEMI-AUTOMATIC DISCHARGE VALVE

Reciprocating compressors usually are of the automatic type; i.e., they feature suction and discharge valves that open and close automatically in response to a positive or negative pressure difference across their ports. Most designs feature a series of reed-type suction valves and one or two disk-type, spring-loaded discharge valves per cylinder. Whereas reed valves have been found to perform satisfactorily in our application, the characteristics of automatic discharge valves lead to an unacceptably low volumetric efficiency, particularly at high compression ratios and when the compressor is not operated at the speed for which the disk valve was optimized. Hence we have decided to retain the concept of automatic valves for the suction valves only, which in our compressor are a series of reed valves arranged around a centrally positioned disk-type discharge valve. The discharge valve should ideally open when, during a compression phase, the pressure in the cylinder reaches the level of the discharge pressure. This event is delayed if the compression ratio is increased, i.e., the crank angle at which equality of pressures is reached on the two sides of the valve disk increases with the compression ratio. Hence we allow the discharge valve to open automatically, but we have developed a concept for its forced closing such that it hits its seat at a selected crank angle; e.g., when the piston just reaches its upper dead center. This new concept of a semi-automatic discharge valve and the mechanism with which it is realized are described in this section.

Concept of a Semi-Automatic Discharge Valve

Closing of the discharge valve is achieved by pushing on its stem with a tappet, whose motion is controlled by a camshaft that is driven off the crankshaft by means of a cog belt. While the tappet is in permanent contact with the camshaft, it touches the valve stem only during, and a very short time after, valve closing. At other times the valve is either in its seat and the tappet above its lower dead position or, during valve opening, the

valve moves toward its open position while the tappet is at its upper dead position. The positions of those extremities of valve stem and tappet that are intermittently in contact with each other are shown as a function of the crank angle in Fig. 4. Notice that the angular velocity of the camshaft is

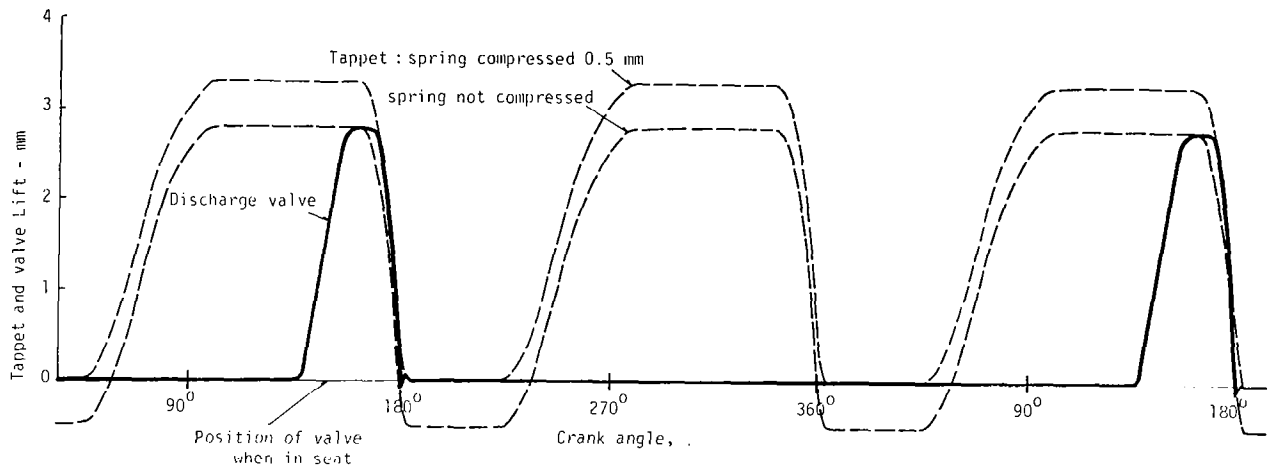


Fig. 4. Position of points on tappet and discharge valve stem, that are intermittently in mutual contact, versus crank angle φ ($\varphi = 180^\circ$: piston is at upper dead center).

twice that of the crankshaft. Only this higher camshaft speed allows achieving sufficiently fast valve closing for the maximum compression ratio of 10 specified by the client. The additional stroke carried out by the tappet between two closing strokes does not have any influence on the discharge valve because the latter is firmly retained in its seat during this phase due to the positive difference between discharge and cylinder pressures. Notice that the tappet's stroke is slightly larger - in our case 0.5 mm - than the valve's stroke. This difference was introduced for reasons of dimensional inaccuracy. One cannot rigidly push with the tappet against the valve while the latter rests in its seat; i.e., one has to incorporate a spring between actual tappet and valve stem as shown in Fig. 5 (see next page). This spring makes it unnecessary to incorporate special damping means for the prevention of too large a jumping back of the discharge valve from its seat. On the other hand, we have incorporated means for the pneumatic damping of the valve when, at the end of the opening phase, it hits its stop. As can be seen in Fig. 5, they consist of a circular groove on the upper side of the valve disk and a fitting counterpart in the structure that provides the guidance for the valve stem.

Mechanism for the Forced Closing of the Discharge Valve

The camshaft/tappet mechanism makes the duration of the valve-closing phase dependent on the compressor speed so that it is always the same fraction of the time of one revolution. Additional measures are necessary to minimize the valve closing in terms of the crank angle interval. Another

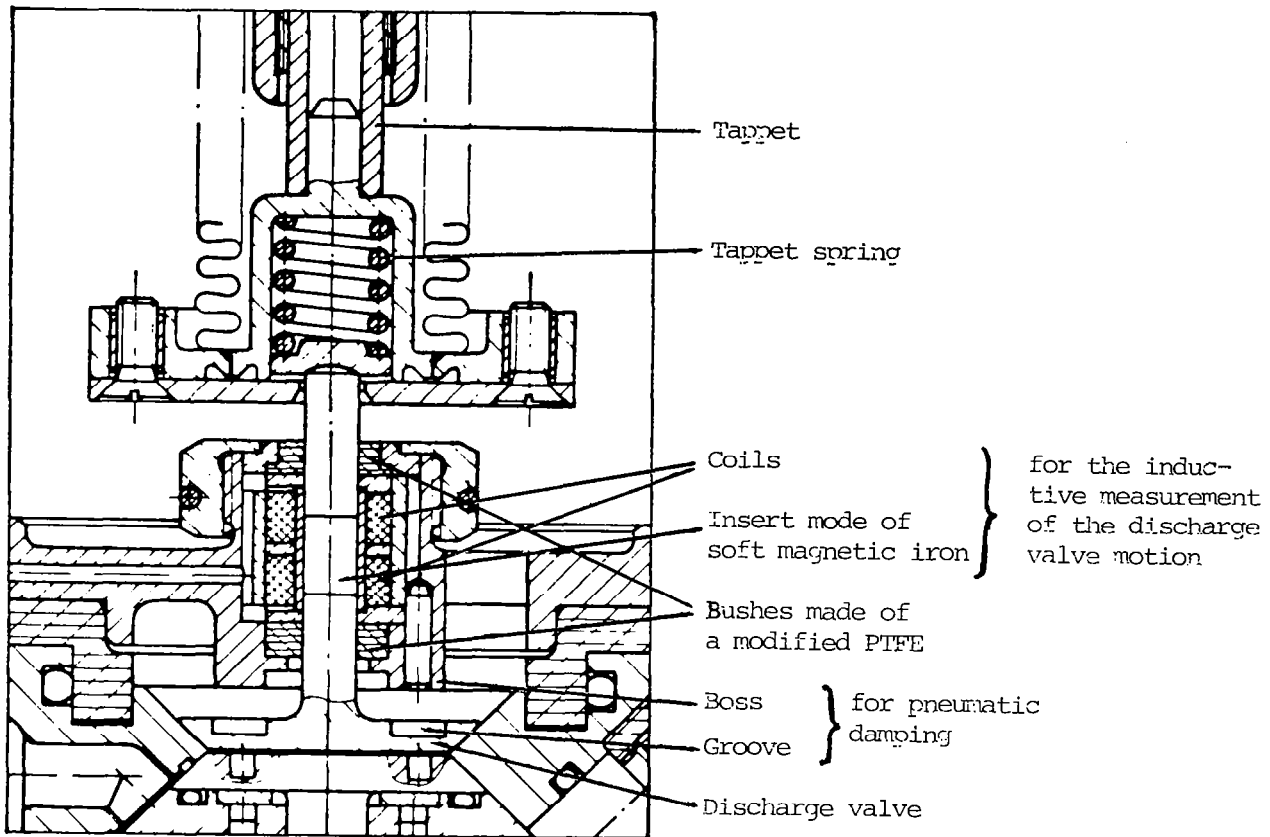


Fig. 5. Details of lower portion of tappet, of discharge valve, and of means for the recording of the valve motion

consideration for the design of the cam contour is the possibly uncontrolled movement of the discharge valve near the end of the closing phase. Ideally, the control mechanism should achieve closing at a very specific crankshaft position. This is not really possible. The tappet, and the discharge valve along with it, is accelerated during most of the closing phase. However, the tappet reaches a position from which it must be decelerated since, slightly later, it must come to a full stop. The valve does not follow this movement; i.e., it remains in contact with the tappet only during the acceleration portion and then proceeds at a constant speed when the tappet is decelerated, meaning that it precedes the tappet from this moment on. Obviously then, a further aim is to postpone the start of tappet deceleration to the last possible moment or, to put it another way, to keep as short as possible the crank angle interval during which the tappet decelerates. There is a limit to the delay of the beginning of the deceleration because the rate of deceleration increases if the deceleration phase is reduced. Noting that during deceleration the tappet roller would jump off the cam if not pushed against it, it is

evident that the larger the deceleration rate, the larger must be the force with which the tappet is pushed against the cam. This force then causes a larger Hertzian surface pressure at all other times and hence an increased wear. Thus a compromise must be found between the duration of the deceleration phase and the mean force with which the tappet is pushed against the cam. To meet the different requirements mentioned above, we had to design a cam of a size that is rather large when compared with the crankshaft [5]. The cam shape is depicted in Fig. 6, which is not to scale but illustrates all relevant features.

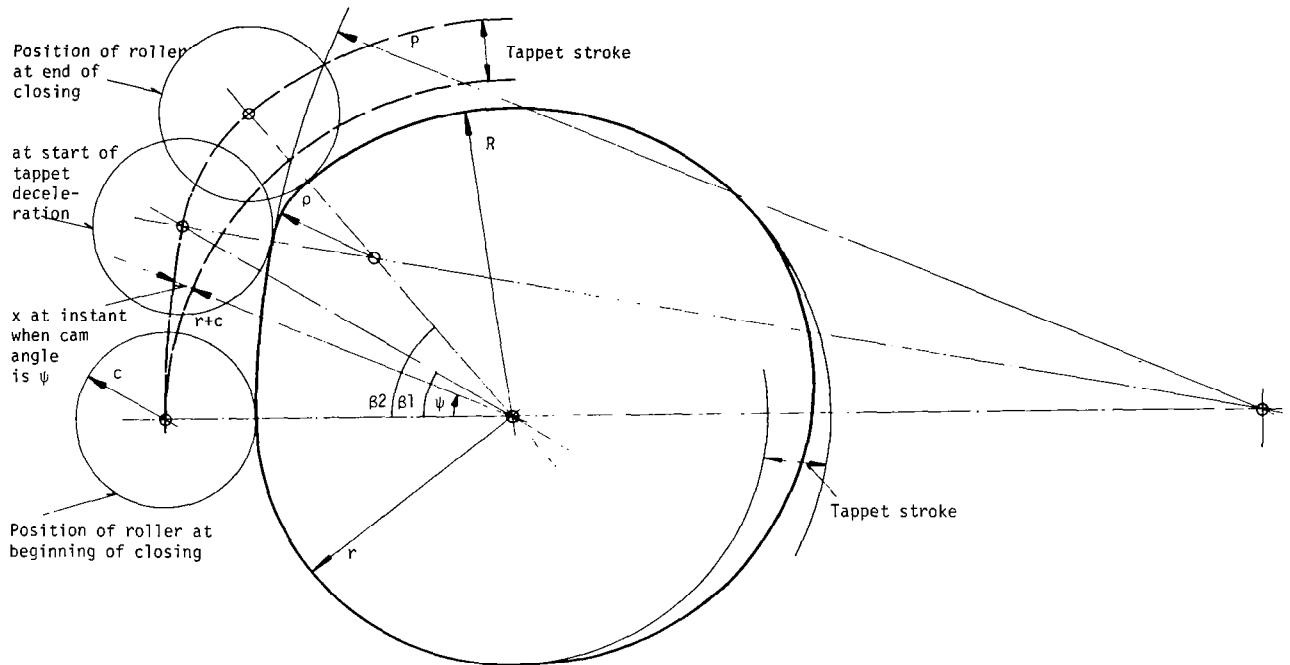


Fig. 6. Cam geometry and motion of the tappet roller over the cam surface during the closing phase.

The position x of the tappet during the closing phase is also shown as a function of the cam angle ψ in Fig. 7 (see next page) together with the tappet speed and acceleration. Also indicated in this diagram are the actual optimized geometrical parameters (in mm) of the cam and the radius of the tappet roller. Notice that the final acceleration at a cam angle $\psi = \beta_1 \approx 35^\circ$ is about 800 m/s^2 . Since the valve has a mass of 5 gm , it resists the push by the tappet spring during final acceleration with about $0.005 \times 800 = 4 \text{ N}$ only. The spring being pre-loaded with about 10 N , it does not seem to be compressed at all at the instant when deceleration starts, and the valve is only about $2.8 \text{ mm} - 2.5 \text{ mm} = 0.3 \text{ mm}$ away from its seat at this instant. In addition, we infer from Fig. 7 that valve closing occurs over a cam angle interval $\Delta\psi \approx 37^\circ$ only; i.e., over slightly more than 5% of one crankshaft revolution.

As noted previously, the discharge manifold is hermetically sealed from the tappet/camshaft assembly by means of a metal bellows. This element has an

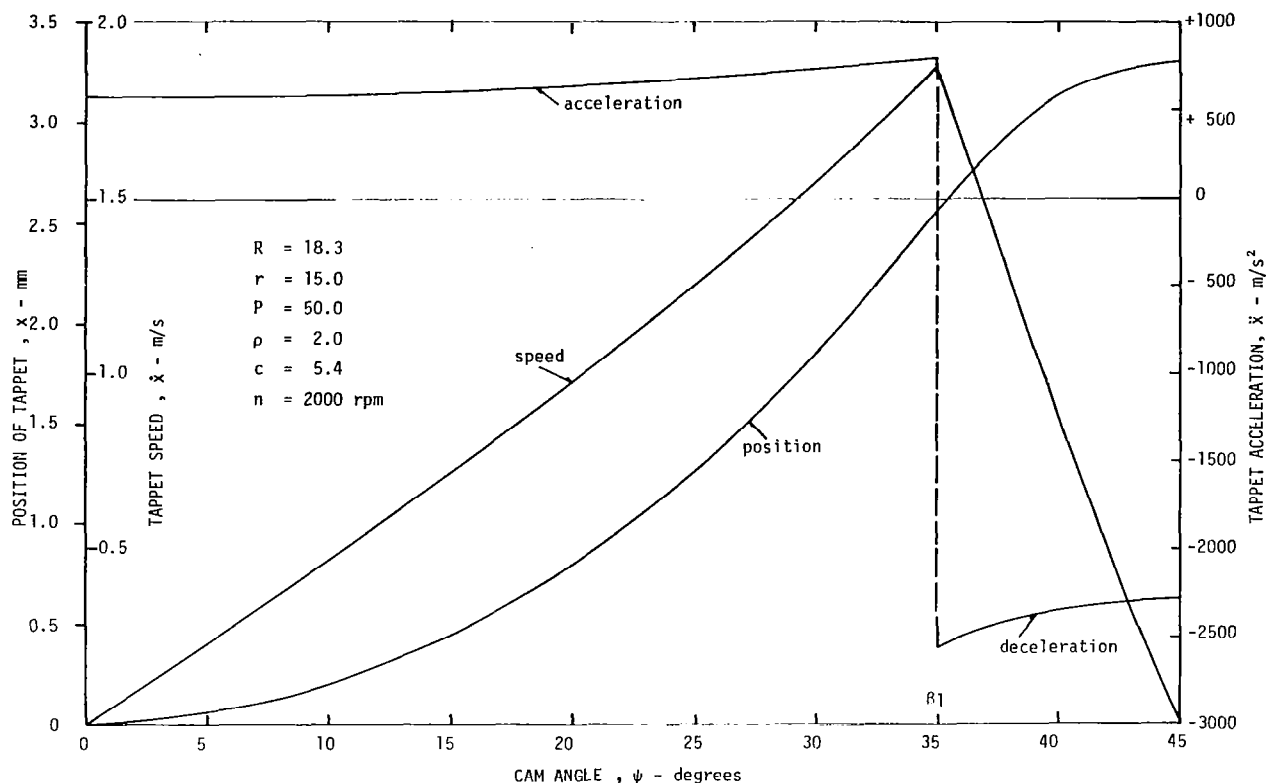


Fig. 7. Characteristics of tappet motion during valve closing.
Camshaft speed: 2000 rpm.

additional function: the pressure in the discharge manifold being typically between 5 and 10 bars higher than in the camshaft housing, the bellows furnishes a pressure force with which the tappet is pushed against the camshaft, thereby assuring a permanent contact between the two elements. With a maximum crankshaft speed of 1000 rpm the bellows is subjected to load cycles at a frequency of 33 Hz. This value is about an order of magnitude above the maximum frequency recommended by the bellows manufacturers. Moreover, the client having specified a minimum operating lifetime of 500 hours at maximum compressor speed, the bellows must survive at least 60 million load cycles, which again is one or two orders of magnitude above manufacturer's specifications. We believe that such a requirement can be met only if, unlike what seems to be general practice, the bellows is deflected only in the strictly elastic range of its material's stress deflection curve [3]. This means that the ratio of the bellows' stroke to its free length has to be limited to a few percent. Stroke and free length are 3.3 mm and 60 mm, respectively, in our design.

Tribological Problems

Because of the small size of the compressor, we had initially designed the tappet as a simple, single element. Friction and wear between tappet and cam were reduced through oil lubrication. Unfortunately, we could not maintain a stable oil film between the gliding elements at high camshaft speeds, and excessive wear resulted under these conditions. Since an effective oil lubrication would be even more difficult to maintain under null gravity condition, owing to the absence of a defined level of the oil pool, we have abandoned the gliding contact with oil lubrication in favor of a practically pure rolling contact between the cam and a roller held in the tappet head with miniature ball bearings. Whereas the ball bearings are grease lubricated for life, there is no lubrication at all between roller and cam. Rather extensive testing with camshafts and rollers of different materials, different surface treatments and/or coatings were required to identify that material combination that resulted in the least wear of the cam and roller surfaces. Generally speaking, the camshaft has been found to be the **critical** element, apparently because of the large radii of curvature of the cam's surface compared with the roller's radius. Specifically, the surfaces of cams made of nitrided steel or coated with tungsten carbide or titanium carbide were found to deteriorate rather rapidly. All roller materials tested performed satisfactorily. These were nitrided steel 34CrAlMo5 and through-hardened steel X102CrMo17 or X165CrMoV12. Excellent results have been obtained with a camshaft made of X102CrMo17 steel, through-hardened and paired with a roller made of the nitrided steel mentioned above. Also very good results were obtained with boronized cam surfaces (material: steel 42CrMo4 boronized to a depth of 100 μm and steel X165CrMoV12 boronized 20 μm deep), paired with rollers made of either one of the three materials mentioned above. With the Hertzian pressure varying between 450 N/mm^2 and 550 N/mm^2 during one revolution of the camshaft, we observed some minor wear of these cams during the first 50 hours of testing at a camshaft speed of 2000 rpm and practically no wear thereafter during several thousand hours of testing.

Guidance of the tappet is provided by a dry bearing material consisting of a layer of sintered bronze that is impregnated with PTFE. The tappet is made of titanium which, according to the dry bearing material's manufacturer, is not hard enough for minimum wear of its material. Therefore we have coated the tappet surfaces with tungsten carbide that is applied with a flame plating process.

Guidance of the discharge valve stem is less critical than tappet guidance because, in contrast to the tappet, the valve is not subjected to significant moments or forces normal to the stem axis. Hence the guiding bushes for the valve stem are made of a modified PTFE only, and the titanium valve is not coated. However, as can be seen in Fig. 5, a piece of soft iron is interposed between upper and lower stem portion. Since titanium is a reasonably good dielectric, **this intermediate stem piece, together with the two coils shown in Fig. 5 inside of a tube made of soft magnetic iron, allows the inductive measurement and recording of the discharge valve motion.** The path of the discharge valve shown in Fig. 4 has been redrawn from such a recording.

SUMMARY

The most important elements of the mechanical design of a reciprocating compressor have been discussed. Since this compressor is a component of a reversed Carnot cycle heat pump that will be used on space missions, its design is strongly influenced by two requirements: the capability of operating in the absence of gravity, and the necessity of achieving as high an efficiency as possible in order to minimize the heat pump's power consumption.

Because of the zero-g environment, we have replaced oil by grease as a lubricant, which made it necessary to strictly separate the heat pump's working fluid from the lubricants. This requirement led to the incorporation of metal bellows with extremely long operating lifetimes.

The concept of a semi-automatic discharge valve was introduced in order to optimize the compressor efficiency. This valve opens automatically when the pressure in the cylinder starts exceeding the pressure in the discharge manifold, but it is closed by means of a mechanism such that complete closing is reached at a particular crank angle or piston position. The main features of this mechanism have been described.

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Mr. Berner emigrated to the United States in 1954, becoming naturalized in 1960. Prior to returning to Switzerland in 1966, he was employed as a principal research scientist at the Avco-Everett Research Laboratory. In this position, he worked primarily on classified projects dealing in the areas of re-entry physics and ICBM technology. Mr. Berner received his degree in Mechanical Engineering from Swiss Federal Institute of Technology, Zurich, Switzerland, in 1952.

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