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Gas Seal Pad With Herringbone-Grooved Rotor—Stiffness and Load Capacity

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Gas Seal Pad With Herringbone-Grooved Rotor—Stiffness and Load Capacity

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ABSTRACT

The principle of herringbone-grooved journal bearings has been applied to the case of a seal disc running under a finger seal pad. The inward pumping action of herringbone grooves on the disc generates load capacity and stiffness to maintain a fluid film and prevent contact of the pad and disc. This mechanism does not depend on a converging film under the pad, such as analyzed in previous works. Analysis shows that significant stiffness and load capacity can be supplied by herringbone grooves. In order for the grooves to be effective, the seal pressure drop must be taken outside of the grooved portion of the rotor, but this may be acceptable in order to gain freedom from maintaining a precise film convergence.

NOMENCLATURE

B	Pad dimension in direction of rotation, m
c	Film thickness over land and in ungrooved region, m
D	Seal diameter (fig. 3b)
h_g	Film thickness over groove, m
H	Film thickness ratio, $h_g/c = 1 + \delta/c$
H_c	Clearance ratio from leading to trailing pad edge
k	Film stiffness, N/m
K	Nondimensional stiffness, kc/p_oLB
L	Pad dimension normal to rotation direction, m
L_1	Total length of pad covered by grooves
p_o	Ambient pressure, N/m^2 abs.
P_s	Pressure ratio across pad normal to direction of rotation
R	Seal radius (fig. 3b)
V	Runner speed, m/s
w	Pad load, N
W	Nondimensional load, w/p_oLB
α	Groove width ratio
β	Groove angle measured from circumferential line
δ	Groove depth, m
θ	Angular coordinate
Λ_b	Compressibility number, $6\mu VB/(p_oc^2)$
μ	Dynamic viscosity, N sec/ m^2

INTRODUCTION

Fluid film slider bearings have been studied for many years, both as self-acting bearings and with external pressurization, where the pressurized lubricant is usually supplied through restrictors in the pad. Recently, Fleming [1, 2] studied the case of a rectangular slider bearing with a pressure flow transverse to the direction of motion. The impetus for this work was a new type of seal, the *padded finger seal* [3]. In this configuration, a seal ring is divided circumferentially into a multitude of segments; each segment, or pad, is supported by a thin sheet metal finger. The concept is illustrated in figure 1 which shows the seal from the downstream side; figure 2 shows a single finger and pad from a different angle. A complete seal has another row of fingers without pads, upstream of the row

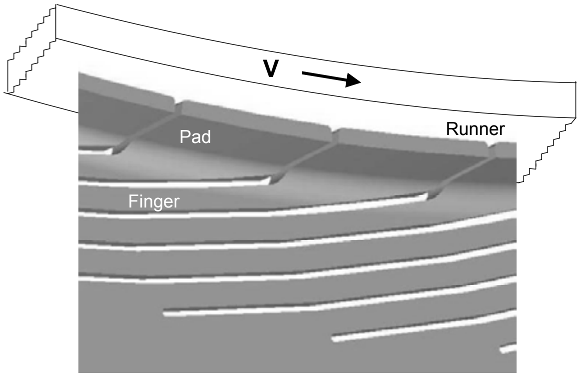


Figure 1. Finger seal concept

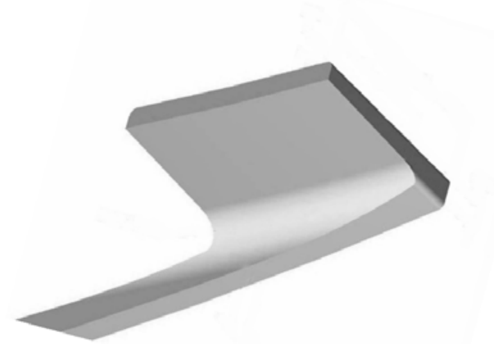


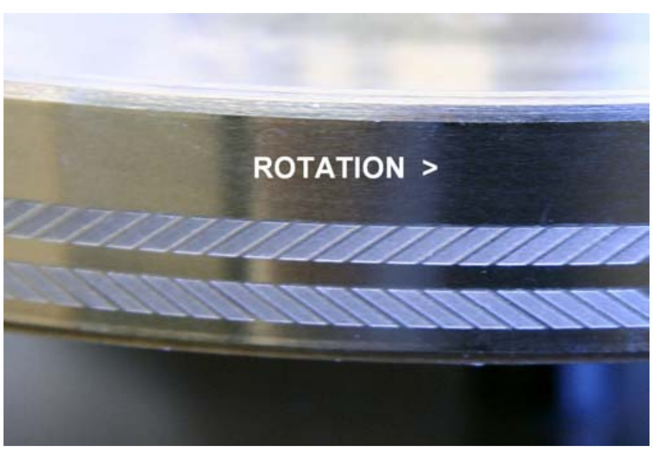
Figure 2. Single pad and finger

shown, arranged to block the leakage flow between the downstream fingers. The intent of the finger seal concept is that the pad will ride on a thin film of fluid while the flexible finger will allow adaptation to shaft vibration or thermal growth. The thin film results in low leakage and also long life, as there is no material contact to cause wear. In operation, the clearance under the individual pad is determined by a force balance between the elastic finger and the fluid film under the pad. Thus one desires a film profile that will allow adequate fluid force to be developed by the pad, such that contact does not occur between the pad and the rotating shaft. Fleming [1] found that load and stiffness can be developed in the fluid film by providing a film profile that converges in the direction of motion; moreover, the film stiffness increases with an increase in the sealed pressure. However, ensuring proper film convergence can be problematic, and it was desired to devise a seal design that did not depend on film convergence.

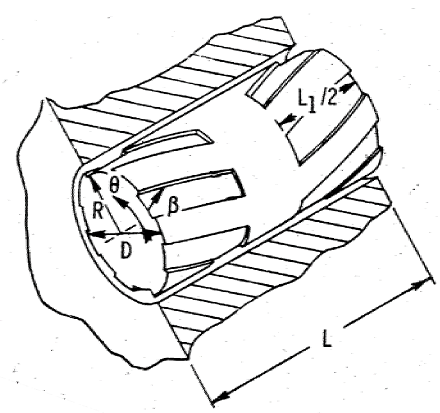
In the field of journal bearings, a bearing capable of carrying high loads is the herringbone grooved bearing [4, 5]. The question arose as to whether a herringbone grooved rotor running against a finger seal pad could generate adequate fluid film load and stiffness to maintain a fluid film. In the present paper, that configuration is analyzed. A herringbone grooved rotor is shown as figure 3. The grooves are angled such that fluid is pumped from the axial edges of the pad to the center.

PROCEDURE

The starting point was the computer code SPIRALG [6], which was written to analyze load, stiffness, and leakage in spiral and herringbone grooved face seals and cylindrical seals. SPIRALG solves the Reynolds equation in the seal using a formulation similar to that set down by Vohr and Chow [4]. The code was modified for this work to analyze a partial arc seal instead of a full circular seal.



(a) grooves on seal disc



(b) nomenclature

Figure 3. Herringbone grooved rotor.

The first finding was that grooves were not beneficial if the pressure drop was taken across the pad. Thus it was assumed for the rest of this work that the pressure drop was taken across a seal dam upstream of the pad being analyzed, and that all edges of the pad were at ambient pressure. With this arrangement, herringbone grooves on the rotor showed promise for a successful finger seal design. SPIRALG was then combined with the optimization code used by Hamrock and Fleming [5] to determine optimum groove parameters.

As in [1], it is convenient to carry out the optimization and present the results in nondimensional form. There are four parameters to be optimized: (1) the film thickness ratio $\boldsymbol{H}$, i.e., the film thickness under the groove divided by the film thickness under the land; (2) the groove width ratio $\boldsymbol{\alpha}$, i.e., the width of the groove divided by the width of the groove-ridge pair; (3) the groove angle $\boldsymbol{\beta}$ (see fig. 3(b)); (4) the groove length ratio $\boldsymbol{L_1/L}$, i.e., the total axial length of the grooves under the pad divided by the axial dimension of the pad. The optimization was carried out two ways: first to maximize pad fluid film load capacity, and second to maximize pad film stiffness; the latter is more important for the finger seal in that it determines whether load capacity can be maintained as the film thickness varies in operation. Three pad aspect ratios were considered: $\boldsymbol{L/B}$ = 0.5, 1, and 2. Results are presented as a function of the nondimensional *compressibility number* $\boldsymbol{\Lambda_b}$.

RESULTS

Figure 4 shows the pad film stiffness achieved for the various cases. The letters *K* and *W* on the curve identifiers in this and subsequent figures indicate whether stiffness or load was maximized for the particular curve. As expected, higher stiffness is produced when that is the parameter being maximized; when pad load is maximized, stiffness is somewhat lower. The differences between the two cases become more pronounced as compressibility number $\boldsymbol{\Lambda_b}$ increases. Stiffness seems to be approaching an asymptotic limit at high compressibility numbers. Additionally, for some cases at intermediate values of $\boldsymbol{\Lambda_b}$, stiffness reaches a peak followed by a decrease at higher $\boldsymbol{\Lambda_b}$. There are substantial differences in stiffness for different pad aspect ratios; lower aspect ratios (relatively greater circumferential dimension) produce greater stiffness.

Figure 5 shows the corresponding loads produced; the loads are naturally higher when load is maximized than when stiffness is maximized. The same trends are seen as for stiffness in that lower aspect ratio pads will carry larger loads. Loads also appear to approach an asymptotic limit at high $\boldsymbol{\Lambda_b}$; this behavior is typical of ungrooved gas bearings, but not of herringbone groove journal bearings [5]. The spread in the curves does not appear as great as in the stiffness plot of figure 4.

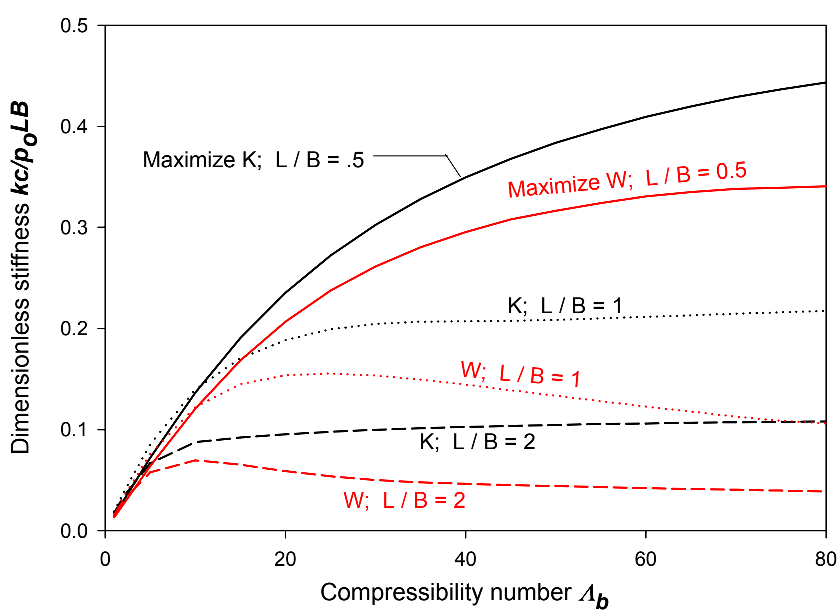


Figure 4. Maximum pad stiffness as a function of compressibility number Λb for three aspect ratios $\boldsymbol{L/B}$

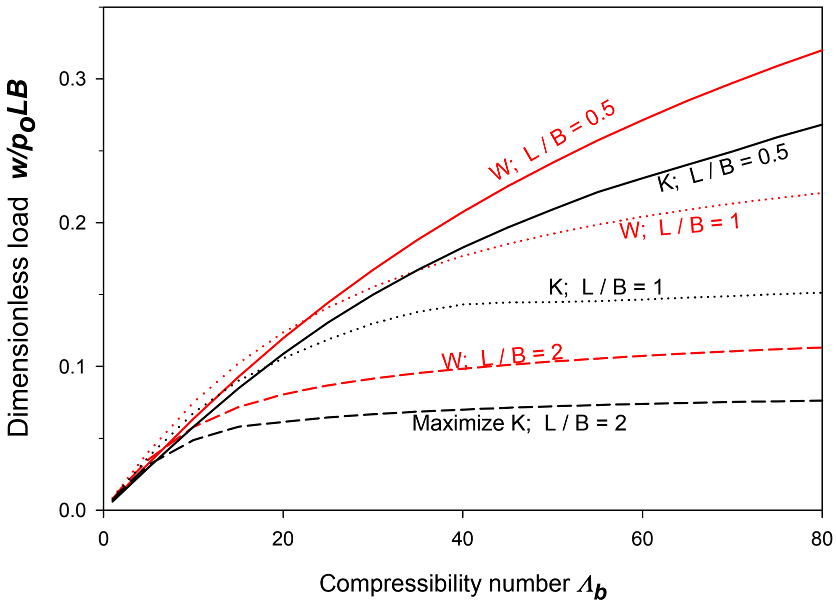


Figure 5. Maximum pad load as function of compressibility number Λ_b for three aspect ratios L / B

Figures 6-9 show values for the optimum groove parameters of film thickness ratio $\boldsymbol{H}$, groove width ratio $\boldsymbol{\alpha}$, groove angle $\boldsymbol{\beta}$, and groove length ratio $\boldsymbol{L_1/L}$, respectively.

Considerably deeper grooves (fig. 6) are required to maximize load than to maximize stiffness. When maximizing load, deeper grooves are needed for larger aspect ratios (smaller circumferential extent of pad) and higher compressibility numbers. For maximum stiffness pads, optimum groove depth does not change much with aspect ratio or Λ_b.

When selecting groove widths to maximize stiffness, the optimization code sometimes called for quite wide grooves ($\boldsymbol{\alpha}$ as much as 0.9). An arbitrary decision was made to limit the groove width ratio to 0.8 to enable practical manufacturing; thus 0.8 is the maximum width shown in figure 7. Computer runs made with unrestricted groove widths showed virtually no change in maximum achievable stiffness, thus there is no practical loss with the restriction.

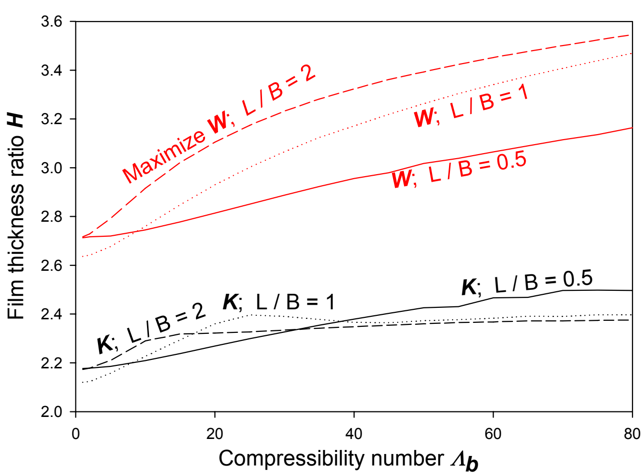


Figure 6. Optimum groove depth ratio $\boldsymbol{H}$ for three pad aspect ratios $\boldsymbol{L / B}$

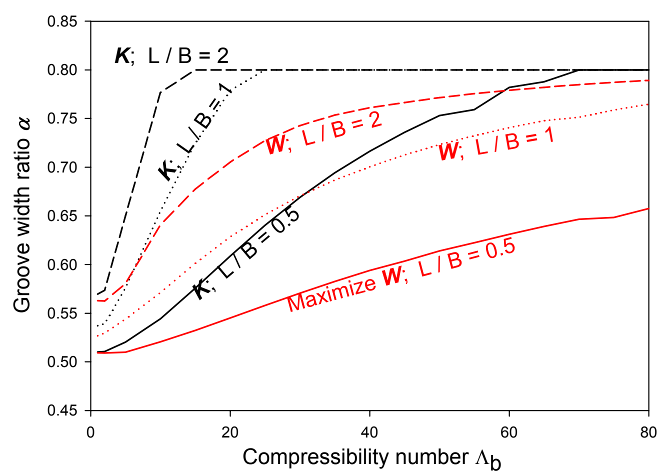


Figure 7. Optimum groove width ratio α for three pad aspect ratios L / B

As figure 8 shows, optimum groove angles, $\boldsymbol{\beta}$, all fall within a 20 degree range. In general, $\boldsymbol{\beta}$ rises with aspect ratio. The exception to this trend is for an aspect ratio of 2 for maximum stiffness, when the optimum groove angle is sometimes lower than for a unity aspect ratio. For the case of $\boldsymbol{L / B}$ = 2, there appeared to be two local optima for higher compressibility

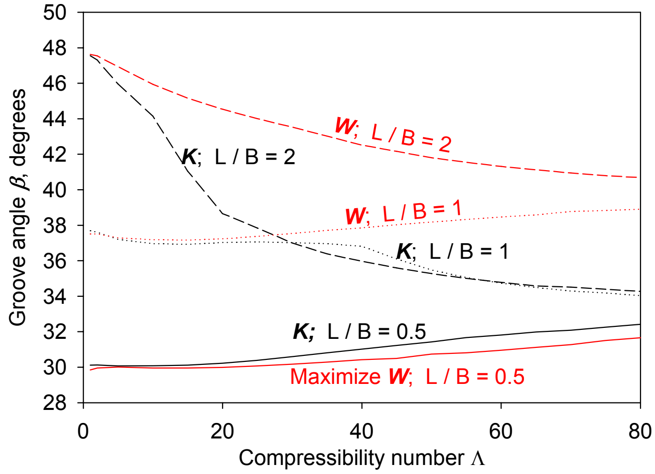


Figure 8. Optimum groove angle β for three pad aspect ratios L / B

numbers. The one reported herein is that which yielded the higher stiffness.

Figure 9 shows optimum groove length ratios. Again, the data for maximizing stiffness at an aspect ratio of 2 are anomalous. The second local optimum, not shown, has groove length ratios in the 0.7 range, yielding a maximum stiffness about 10 percent less than given in figure 4.

It is appropriate at this point to make a comparison with the previous work on ungrooved, convergent-film pads, reported in [1]. Stiffness and load capacity are shown in figures 10 and 11, respectively, for a square pad (aspect ratio L / B = 1) with various degrees of convergence (H_c) and various pressure ratios (P_s) across the pad.

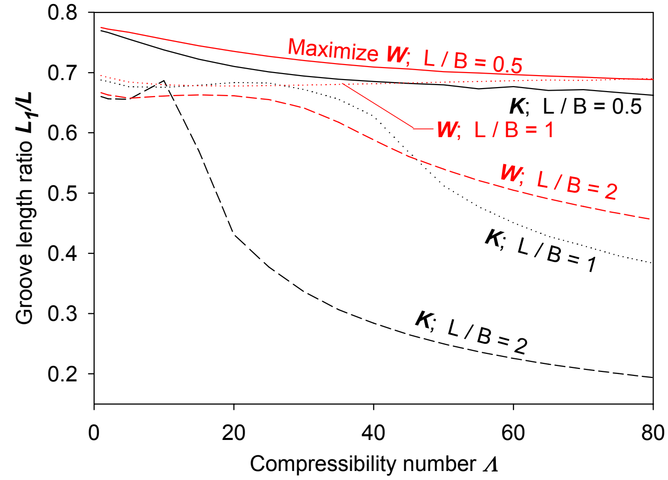


Figure 9. Optimum groove length ratio L_1/L for three pad aspect ratios

The pressure ratio corresponding to that assumed in the present work is one. Convergence H_c is the ratio of leading edge to trailing edge clearance; the convergence of the pads in the present work is 1. Pads are compared for the same minimum clearance (i.e., trailing edge clearance for a convergent-film pad.

Maximum stiffness data from figure 4 for a square pad (L/B = 1) have been added to figure 10 as a dash-double dot curve. One sees that, for a pressure ratio of 1, a herringbone grooved rotor with a nonconvergent pad generates lower stiffness than a plain rotor - convergent pad combination with either of the convergence ratios presented. For a pressure ratio of 5 and a convergence of 4, the convergent pad stiffness is more than double that of the herringbone pad at high compressibility numbers.

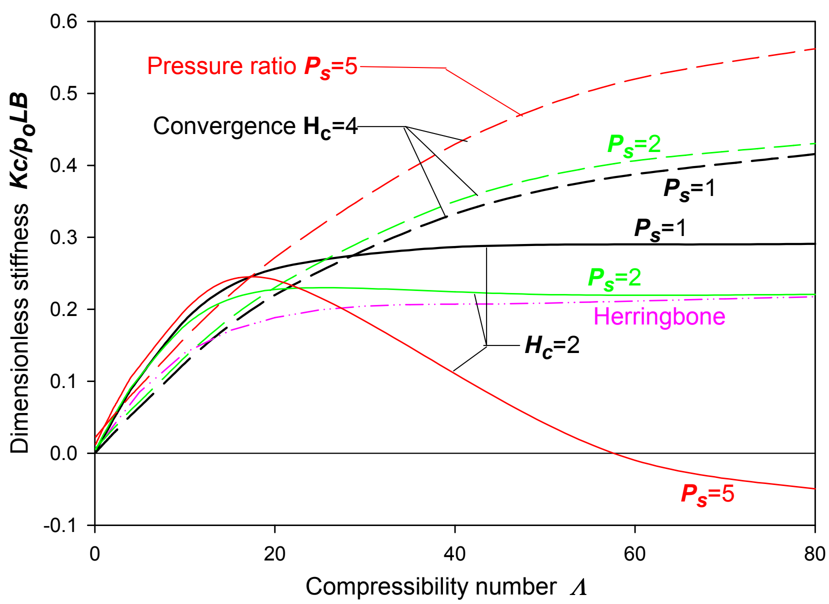


Figure 10. Stiffness for square convergent-film pad on smooth rotor (from [1]) and herringbone rotor with nonconvergent film.

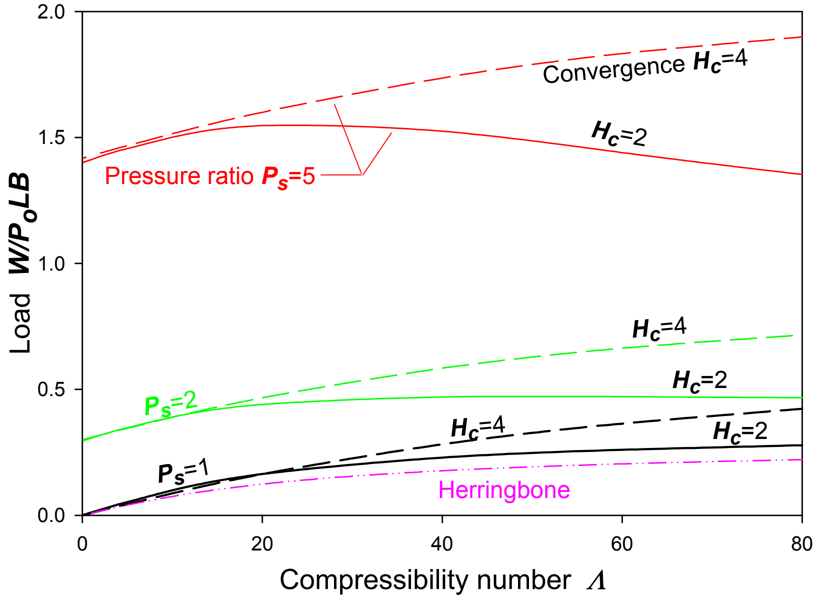


Figure 11. Load for square convergent-film pad (from [1]) and herringbone rotor with nonconvergent film.

As for load, maximum load data from figures 5 have been placed in figure 11 as a dash-double dot curve. Similar to the results for stiffness, loads generated by the herringbone arrangement are somewhat less than those generated by the plain rotor – convergent pad combination for a pressure ratio of 1 and convergence of 2. For greater convergence or pressure ratios above 1, the load carried by the convergent pad is much larger than for the herringbone configuration. As was stated, however, for the intended finger seal application, stiffness is a more important property than load. Depending on the particular seal design, the herringbone configuration may provide adequate performance. Although not shown, limited studies indicated that tilting a pad (to make a convergent film) facing a herringbone-grooved rotor increased the load and stiffness over the case with grooves alone; however, the tilted pad without grooves produced still greater load and stiffness.

CONCLUDING REMARKS

Analysis shows that significant film stiffness and load capacity can be supplied by herringbone grooves under a finger seal pad even when the pad is untilted. Groove parameters were optimized to obtain either maximum load capacity or maximum stiffness. Although effective, load and stiffness for the grooved case were somewhat less than for pad tilt alone. Also, in order for the grooves to be effective, the seal pressure drop must be taken outside of the grooved portion of the rotor, but this may be acceptable in order to gain freedom from precise pad tilt.

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13. ABSTRACT ***(Maximum 200 words)***

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