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Approximate Solution for Choked Flow in Gas Seal Pads

David P. Fleming
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February 2004

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APPROXIMATE SOLUTION FOR CHOKED FLOW IN GAS SEAL PADS

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ABSTRACT

Previous analyses of high pressure seals have considered adiabatic flow with friction but neglected effects of seal rotation. Most of this work analyzed a one-dimensional flow field. This works well to calculate stiffness and leakage of full circular seals, either face seals or annular ring seals. However, it cannot provide accurate results for a rectangular seal pad with its strongly two-dimensional flow field and its reliance on hydrodynamic forces to maintain a full fluid film. On the other hand, solutions of Reynolds lubrication equation have been obtained for the two-dimensional flow in a seal pad. But these solutions do not account for choking which occurs at high seal pressure ratios, nor do they consider the pressure loss that occurs in the entrance region of the flow field. The aim of the present work is to build on the Reynolds equation solution by use of an approximate choked flow analysis. This will account for the pressure losses in the flow entrance region, ensure that fluid velocities remain subsonic, and enable fluid inertial effects within the pad film to be accounted for. Results show that, in general, fluid inertia acts to decrease pad film load capacity and leakage, and increase film stiffness.

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. In more recent years, the bearing properties of ring seals have been studied, e.g. by Black (1969), wherein stiffness and damping are developed by the fluid flowing through the seal clearance. Analyses have been conducted which show that substantial stiffness can be developed with either a liquid or a gas as the sealed fluid (e.g., Fleming, 1977, 1979). Maximum stiffness is developed when the film thickness decreases in the flow direction, either by a discrete step or by a continuous taper.

A new type of low-leakage seal, the *padded finger seal*, has been proposed (Braun et al., 2002). In this configuration, the 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 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 between

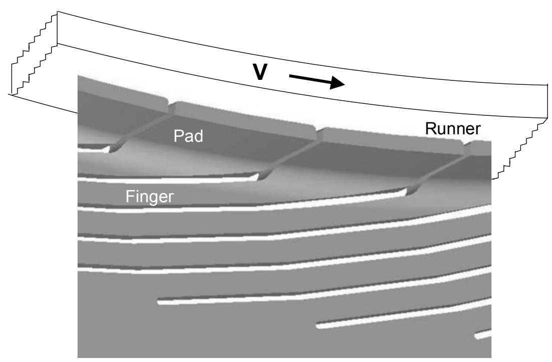

Figure 1. Finger seal concept

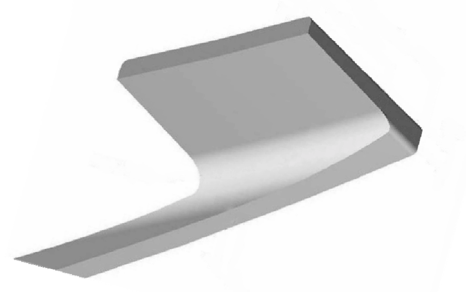

Figure 2. Single pad and finger

the pad and runner. Thus one desires a film profile that will allow adequate fluid force to be developed such that contact does not occur between pad and runner.

As the circumferential extent of each pad is small (6 degrees or less), the pad can be treated as a rectangular slider bearing. However, in contrast to the classical slider, there is a pressure difference from the upstream to the downstream side of the pad which is likely to have a substantial effect on the load capacity of the pad. Figure 3 shows the geometry of the single pad to be analyzed. It depicts a plane surface rectangular pad with a portion of a runner positioned over it. As shown, the fluid film may converge in both the velocity and flow directions. Vertical dimensions are exaggerated for clarity. The pressures are shown as p_s on the upstream edge of the pad and p_o on the downstream edge and sides. In a previous paper by the author (Fleming, 2003), properties of this gas seal pad were determined with the computer code GCYLT (Gas-lubricated Cylindrical Seals, Turbulent) (Shapiro, 1995) which solves the Reynolds lubrication equation. This solution, however, neglected the effects of fluid inertia, the chief manifestations of which are a pressure drop at the seal entrance and choking at the seal exit where the fluid velocity may approach the speed of sound. The object of this paper is to account for fluid inertia and determine its effects on seal properties.

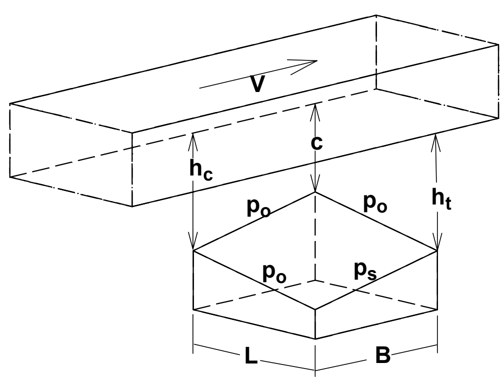

Figure 3. Pad and runner schematic

NOMENCLATURE

B Pad dimension in direction of motion, m

c Minimum film thickness, m

h Local film thickness, m

h_c Maximum film thickness on line in direction of motion from where $h = c$, m

h_t Maximum film thickness on line normal to motion direction from where $h = c$, m

k Film stiffness, N/m

L Pad dimension normal to motion, m

m Leakage rate, kg/s

p_o Downstream pressure, N/m^2 abs.

p_s Upstream pressure, N/m^2 abs.

p_1 Pressure at seal entrance (p_s minus entrance loss)

p_2 Pressure at seal exit (greater than p_o if flow is choked)

P_s Seal pressure ratio p_s / p_o

V Runner speed, m/s

W Pad load, N

PROCEDURE

The computer code GCYLT (Shapiro, 1995) is an ideal tool for analyzing the finger seal pad as long as fluid inertia is not significant. One simply inputs pad geometry and operating conditions, and the code calculates load capacity, stiffness, leakage, and power loss. The code uses a finite-difference scheme to solve the compressible Reynolds equation on a rectangular grid. When fluid inertia is significant, however, the Reynolds equation solution is inadequate; for even moderate seal pressure drop, supersonic velocities are predicted.

For ring seals, where the flow can be assumed one-dimensional, a solution for adiabatic flow with friction may be obtained (Fleming, 1979) starting from the Mach number equation as derived by, e.g., Shapiro (1953). This solution does account for fluid inertia. For the finger seal pad considered herein, however, one-dimensional flow solutions are clearly inadequate. Classical compressible flow texts (e.g., Shapiro, 1953) go only as far as treating isentropic flow in two dimensions. While a two-dimensional numerical solution of the Navier-Stokes equations could no doubt be obtained, the effort required is formidable.

An approximation to the inertial flow may be obtained as follows. The fluid is assumed to flow through the seal clearance in multiple stream tubes arranged in a fan pattern, as depicted in figure 4. In each stream tube, the width of the tube varies with flow distance as shown in figure 4. The film thickness also varies as illustrated in figure 5; the exit clearance can be more or less than the entrance clearance. Figure 5 shows different pressures within and outside the seal passage at the stream tube ends. At the entrance, the pressure drops from p_s to p_1 as the fluid accelerates from upstream stagnation conditions. At the downstream end, p_2 will be greater than the downstream reservoir pressure p_o when the exit is choked, that is, when the exit velocity is sonic.

Fluid pressures and velocities are determined along the stream tube using the computer code AREAX (Zuk and Smith, 1974), which obtains a quasi-one-dimensional solution of the Mach number equation derived by Shapiro (1953). This solution is obtained for

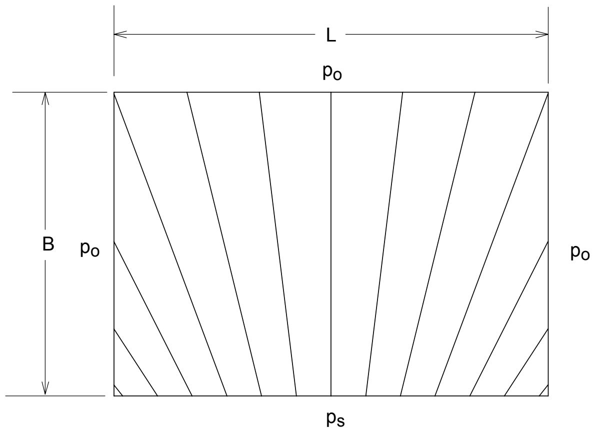

Figure 4. Layout of stream tubes

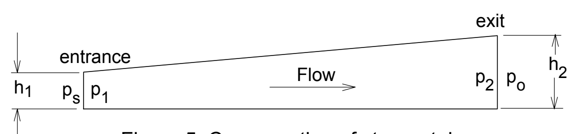

Figure 5. Cross section of stream tube

a stationary seal runner. Code output includes the Mach number distribution along the stream tube as well as entrance and exit pressures (which may be different than the pad boundary conditions shown in figure 3 because of entrance region pressure drop and choking, respectively). These intermediate results calculated by AREAX are then mapped to the rectangular grid used by GCYLT; entrance and exit pressures from AREAX become the boundary pressures for GCYLT, while the interior Mach number distribution is used in the calculation of inertial pressure changes in the pad interior. GCYLT is then run to obtain the final pressure distribution over the pad, taking into account the runner velocity. GCYLT was modified to include the effect of fluid inertia on pressure change between grid points.

For the present work, a uniform grid was used in GCYLT consisting of 29 nodes in the flow direction and 51 nodes in the direction of motion. While this number of points may be more than needed for most operating conditions, it was convenient to use a constant number for all data runs. A slight inaccuracy is produced because the edge pressures are taken as the downstream value on three sides of the pad with the upstream supply pressure on the fourth side; thus there is a large pressure gradient between nodes where the two pressures meet. However, Fleming (2003) found that overall results for the non-inertial solution were little different than for a more accurate modeling arrangement. The code sets the number of stream tubes (fig. 4) for the call to AREAX at 2 less than the number of motion direction grid points, or 49.

Both GCYLT and AREAX normally include effects of turbulence if the Reynolds numbers of the seal flow warrant it. For simplicity, however, the results presented below are for laminar flow.

RESULTS

The new analysis was run for the same design example used for the non-inertial flow analysis (Fleming, 2003). The pad was square with $L = B =$ 7.5 mm and a design-point minimum clearance c of 0.01 mm. The sealed fluid was air at 600 °C with two values of upstream pressure p_s, namely 2 and 5 atmospheres absolute. Downstream pressure p_o was 1 atmosphere; thus pressure ratio across the pad was either 2 or 5.

With the consideration of Mach numbers and possible choked flow, seal performance results do not lend themselves to presentation by non-dimensional variables as was done previously. Therefore all results are dimensional.

Pad load capacity and stiffness

First, the case of film taper in the flow direction was explored to determine whether this alone would be adequate for pad support. Figure 6 shows pad load as a function of minimum clearance for a pressure ratio of 5 and runner speed of 400 m/s. Three cases are shown: convergence only in the motion direction of 0.025 mm, taper in the axial direction of 0.025 mm, and with both convergence and taper. (As the terms are used herein, *convergence* means a reduction in film thickness in the direction of runner motion, while *taper* means a reduction in film thickness in the flow direction.) As figure 6 shows, the load carried by the pad is comparable for the three cases. For the two cases having convergence, however, load decreases

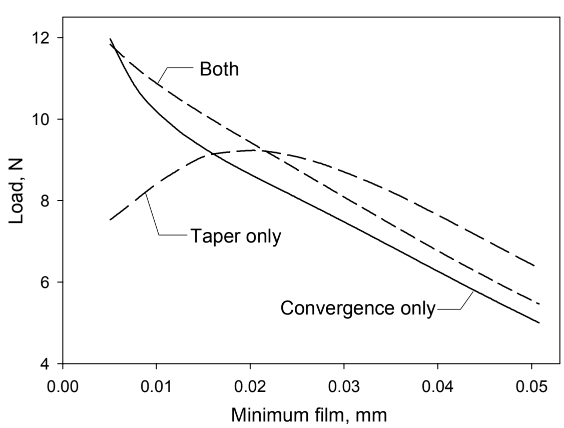

Figure 6. Pad load for convergence, taper, and both. P_S=5, V=400 m/s.

monotonically as film thickness increases; this implies a positive stiffness which is essential for proper operation of the finger seal. For the case of taper only, load increases with film thickness for small values of the latter. This implies negative stiffness in this area, with the potential of contact between pad and runner, an unacceptable situation. Similar results were found for the non-inertial solution. Thus the rest of the results presented will be for convergence in the motion direction only.

Figure 7 compares pad load capacity results for inertial and non-inertial solutions at pressure ratios of 2 and 5, and a runner speed of 400 m/s. The convergence in the motion direction is 0.025 mm. As expected, the pad load is higher for the higher pressure ratio. For a pressure ratio of 5, predicted load is notably lower for the inertial solution at clearances greater than 0.02 mm, while for a pressure ratio of 2, there is little difference between inertial and noninertial solutions. For all results, load drops with increasing clearance, but it drops more for the inertial solution; the non-inertial load becomes nearly constant at large clearances. Thus it is apparent that the inertial solution predicts higher pad stiffness at large clearances. Two aspects of the inertial solution influence these results. First, when the fluid enters the seal clearance it accelerates from the assumed upstream stagnation condition with a resulting pressure drop. This makes the predicted pressure in the film lower than calculated by the non-inertial solution, with a resultant lower load. Second, flow may be choked at the film exit. This means the exit pressure will be greater than the downstream reservoir pressure; film pressures would thus be higher than for the non-inertial solution resulting in a higher load. One or the other aspect may produce a greater effect for a given clearance. For both pressure ratios, an entrance pressure drop ensues. Choking can occur for a pressure ratio of 5, but is unlikely for a pressure ratio of 2 since this is only slightly greater than the critical pressure ratio for air of 1.89.

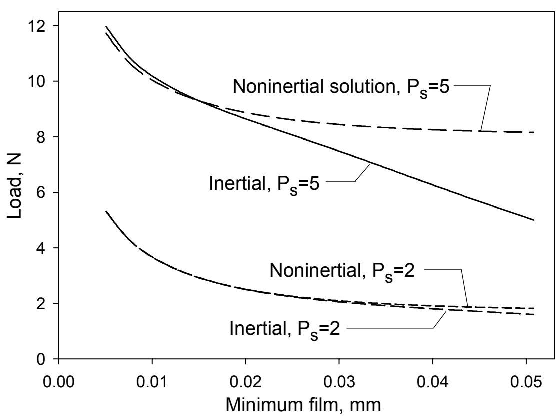

Figure 7. Pad load; V = 400 m/s

For the lower pressure ratio, flow rates are lower, resulting in less entrance loss. This is illustrated in figure 8, which depicts results from the AREAX part of the solution (and thus does not account for runner speed). Entrance pressure drop increases rapidly with clearance. This indicates a greater influence of inertial effects for larger clearances; at the largest clearance of figure 8 the entrance pressure drop exceeds 30 percent of the total sealed pressure. Figure 7 for pad load hints at this, in that there is a greater difference between inertial and non-inertial solutions at larger clearances.

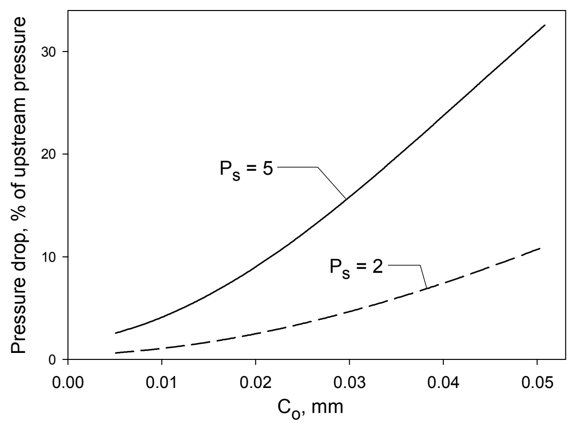

Figure 8. Entrance pressure drop; pressure ratio Ps=5 and 2

Pad stiffness, or the incremental change in load as clearance changes, is shown in figure 9. As was inferred from figure 7, the inertial solution predicts a larger stiffness over most of the clearance range. Fluid inertia has a much larger effect at larger clearance and higher pressure ratio. For a pressure ratio of 5, increasing inertia effects make the stiffness nearly constant at high clearance. For a pressure ratio of 2, inertia effects are much smaller; even so, the stiffness difference between inertial and non-inertial solutions is larger than would be guessed from figure 7. However, the use of a log scale in figure 9 exaggerates the stiffness differences at low stiffness values. For the non-inertial solution, stiffness is virtually unaffected by pressure ratio.

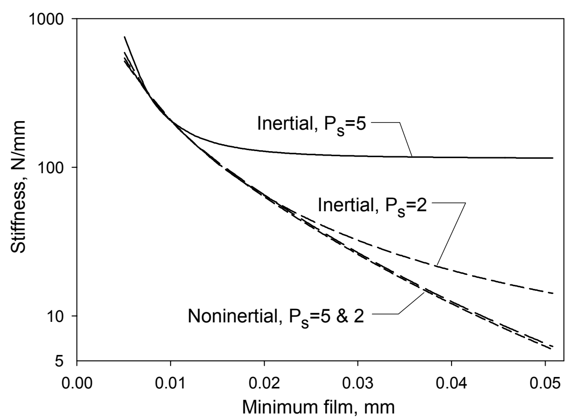

Figure 9. Pad stiffness; V = 400 m/s

The effect of runner speed is shown in figure 10 for a pressure ratio of 5 and speeds of 400, 100, and 10 m/s. The six curves divide into two distinct groups, inertial and noninertial, at larger clearances. In each group the loads are higher for higher speeds. As also seen in figure 7, the non-inertial solution predicts higher loads; moreover, these loads become nearly constant as clearance increases. For the inertial

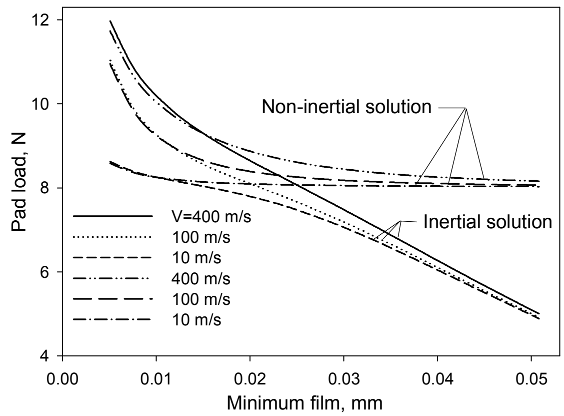

Figure 10. Pad load for various speeds; pressure ratio P_s = 5.

solution, in contrast, load decreases with increasing clearance out to the highest clearance considered. This results in a significantly larger stiffness than for the non-inertial solution, and is a major beneficial effect of fluid inertia. Figure 10 also shows that fluid inertia acts to maintain the stiffness even at the low runner speed of 10 m/s.

Leakage

Up to this point, leakage through the pad film has not been considered. However, as the intended application is a seal, leakage is an important factor. Figure 11 shows the well-known result that that leakage increases very rapidly as clearance and upstream pressure increase; note that the ordinate of figure 11 uses a log scale. Figure 11 is for a speed of 100 m/s; leakage is slightly lower for a speed of 400 m/s. The effect of fluid inertia is marked: compared to the non-inertial solution, leakage considering fluid inertia is much less. Both entrance loss and choking contribute to

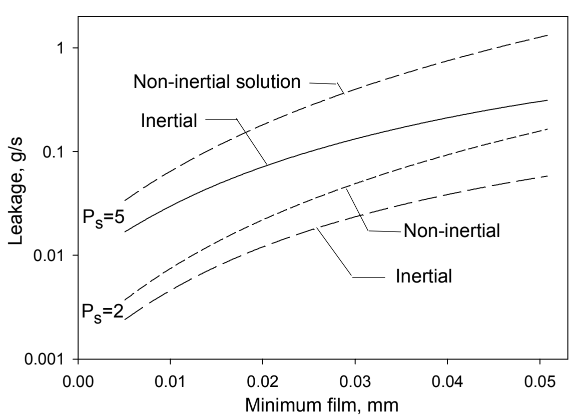

Figure 11. Seal pad leakage; pressure ratios of 5 and 2; V=100 m/s.

this by producing pressure drops at the seal entrance and exit; thus there is a smaller pressure difference to be taken through the fluid film. One may note in this connection that entrance loss also acts to reduce leakage for an incompressible fluid, but of course choking only occurs when the fluid is compressible.

Quality of approximate solution

One measure of the stream tube approximation may be had by comparing the results for load capacity and leakage calculated by AREAX alone with the final GCYLT solution considering inertia. Recall that the stream tube approximation assumes that fluid flows only along the straight stream tubes depicted in figure 4, and this approximation is used to calculate boundary pressures and Mach numbers within the seal pad. If the approximation is accurate, the final GCYLT results for low runner speed should be identical to those calculated by AREAX. In reality, the stream tubes will have some curvature, and the AREAX and GCYLT results will differ.

Figure 12 compares load capacity results from the intermediate solution of AREAX with the final solution from GCYLT. The AREAX solution does not consider runner speed; thus to make the comparison valid,

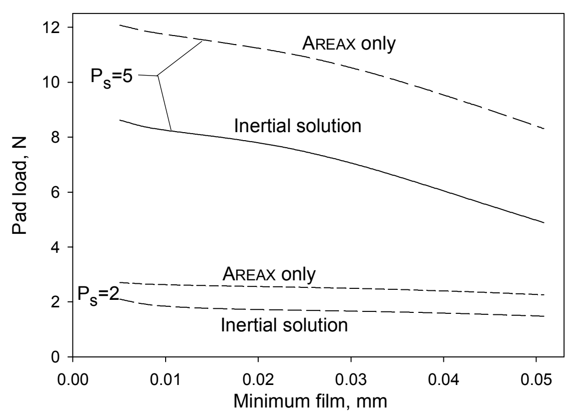

Figure 12. Pad load for full inertial solution and AREAX only; Ps = 5 and 2

GCYLT was run for the low speed of 10 m/s (GCYLT is not structured to work at zero speed). The GCYLT solution predicts pad loads about 30 percent less than the AREAX-only solution; this is approximately the same for both pressure ratios presented in figure 12. A visual estimation of the slopes of the curves, however, indicates that pad stiffness is similar for intermediate and final solutions. The data in figure 13 show that the GCYLT solution predicts 50-60 percent higher leakage than does the AREAX solution; this is consistent with the results for load capacity. Both of these findings indicate that the straight stream tube approximation depicted in figure 3 is not totally accurate; in actuality, the stream tubes are curved in a way that allows greater leakage flow and concomitantly lower load capacity. Thus the present work is only a first step in calculating the properties of seal pads. However, physical reasoning suggests that more accurate analyses will uphold the character of the results obtained herein.

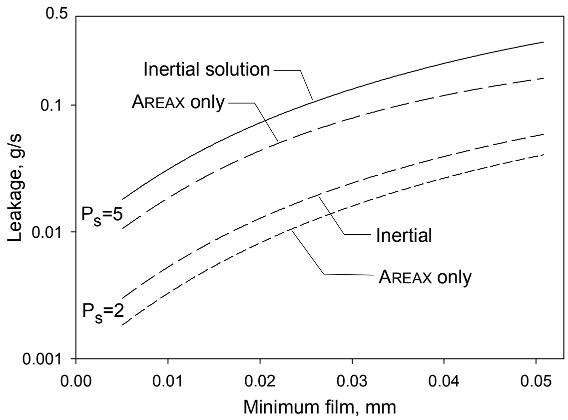

Figure 13. Leakage for full inertial solution and AREAX only; Ps = 5 and 2.

CONCLUDING REMARKS

An approximate solution has been presented to account for fluid inertia in gas flow through rectangular seal pads, where the direction of flow is transverse to the runner velocity. Results show beneficial effects of fluid inertia for the intended finger seal application: generally higher film stiffness and lower leakage compared to a non-inertial solution. Some inaccuracy appears to exist in the solution; however, the results should still prove useful.

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

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