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2002

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ABSTRACT

This paper presents parametric and optmisation studies for a rotaprocating compressor. The effects of various motion profiles, number of cycle per motor revolution and other design parameters were analysed and discussed. The simulation model of the compressor was linked with the Complex optmisation algorithm to search for the combination of design variables that produces an optimum compressor performance. Results show that performance improvement of 10% may be obtained.

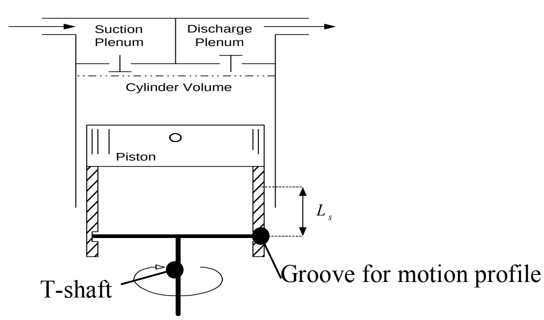
List of Symbols

D	Cylinder Bore, (m)	N	Compressor cycles per motor revolution
a	Acceleration, (m/s^2)	r	Profile pitch radius, (m)
f_1, f_2	Coefficient of friction	α	Pressure angle, (rad)
h	Stroke length, (m)	β_f	Angular displacement for frictional force, (rad)
F_f	Frictional force along piston internal groove surface, (N)	β_m	Angular displacement for driven shaft force, (rad)
F_m	Shaft driven force, (N)	β_n	Angular displacement for normal contact force, (rad)
F_p	Gas pressure load on piston face area, (N)	θ	Angular displacement of crank from TDC, (º)
m	mass of piston, (kg)	ω	Motor's angular velocity, (rad/s)

INTRODUCTION

In previous work [1], a new mechanism has been proposed to integrate the merits of **rota**ry and reci**procating** compressor to result in a new compressor namely **rotaprocating** compressor. This compressor incorporates a new mechanism that replaces the crank and connecting-rod mechanism in the reciprocating compressor and resulted in a compressor that is constructed like a rotary compressor, but works like a reciprocating compressor, see Figure 1. In its simplest form, a rotaprocating mechanism compromises of two main parts: A piston with a groove cut in its internal surface and a T- shaft with the driven end connected to the motor shaft.

Figure 1: Schematic of a rotaprocating compressor.



The driving end is confined in the groove on the piston inner surface. During an operation, the motor shaft rotates the T-shaft and since the piston is restrained from rotation, a linear displacement of the piston is resulted. The capacity of the compressor may increase by multiple folds, even operating at the same compressor speed, if the motion profile is designed to have multiple compressor cycles per motor revolution. However, it was found that the torque required by the compressor was somewhat larger [1] as compared to the reciprocating counter part. Parametric studies have been carried out to further understand the behaviour of the machine and optmisation studies were performed to obtain the combination of design dimensions that produces optimum compressor performance.

MOTION PROFILES

The mathematical model has been developed [1, 2]. It describes the kinematics, thermodynamic, dynamic, fluid flow and heat transfer of the compressor and will not be shown here. In this paper, only the part on the motion profile will be briefly shown and discussed. In general, the design seeks to find a motion profile that has lower velocity, acceleration and jerk, which gives lower driving required. Many of the popular standard cam laws and their composites known as Sine Constant Cosine Acceleration (SCCA) have been examined. The compressor's displacement, velocity, acceleration and jerk and pressure angles for both rise and return cycle employing various motion profiles have been studied. Some of the characteristics are shown in figures 2 to 4.

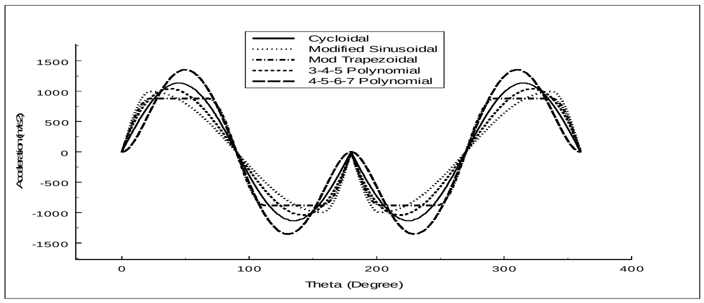


Figure 2: Variation of acceleration

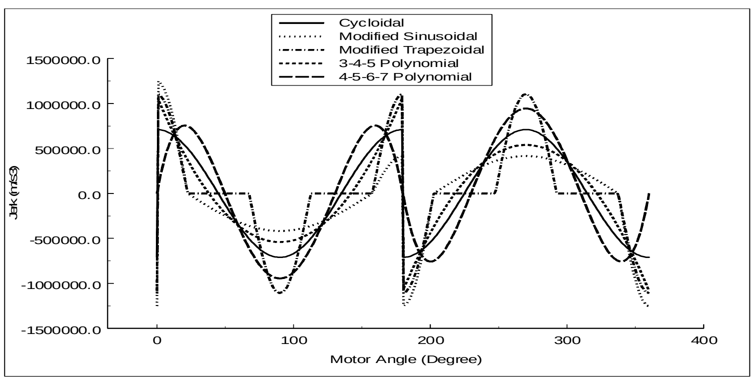


Figure 3: Variation of Jerk

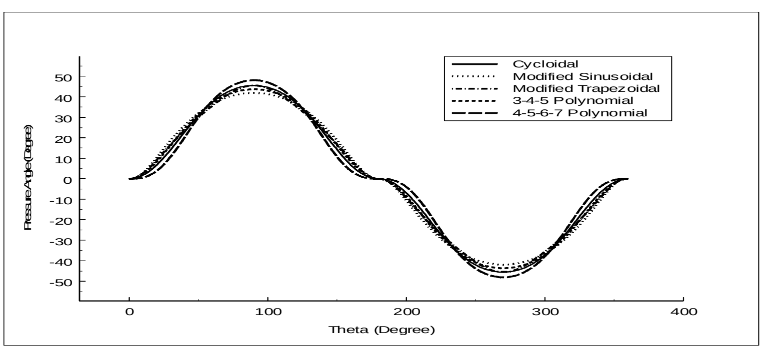


Figure 4: Variation of pressure angle

After considering the kinematics and the mechanical characteristic of the profiles, the modified sinusoidal profile was selected.

This modified sinusoidal is a combination of cycloid and harmonic quadrants occupying different parts of the working period. The equation for piston displacement y in modified sinusoidal motion for rise cycle is given as

$$y = \frac{h}{\pi + 4}\left[\pi \frac{\vartheta}{\beta} - \frac{1}{4}\sin\left(4\pi \frac{\vartheta}{\beta}\right)\right] \qquad 0 \le \vartheta \le \beta \tag{1}$$

The choice of the motion profile has great impacts in the performance of the compressor. This can be seen in the torque equation. The equation for the overall torque of the input shaft [1] is given as

$$T = F_m.2r = \frac{-2r(ma + F_c - F_p)(\cos\beta_n + f_1\cos\beta_f)}{\cos\beta_m\left[2(\sin\beta_n + f_1\sin\beta_f) \mp f_2\left|\frac{-2r(\cos\beta_n + f_1\cos\beta_f)}{(r+l)\cos\beta_m}\right|\right]} \tag{2}$$

where $\beta_f, \beta_m, \beta_n$ are functions of pressure angle and will take an appropriate form depending on the position of the piston [1]. The pressure angle can be seen as the slope of the motion profile and in general, higher pressure angles resulted in higher torque [1].

PARAMETRIC STUDIES

For the present study, the initial compressor specifications and the operating conditions are shown in Table 1. In this section, the effects of various design parameters on compressor performance are studied.

Table 1: Initial compressor specifications and operating conditions

Compressor Specifications	
Stroke length (mm)	18.0
Swept volume (m^3)	9.1207×10^{-6}
Clearance length (% of swept volume)	1
Compressor cycle per motor revolution	1
Piston diameter (mm)	25.4
Profile pitch diameter (mm)	22.5
Suction port diameter (mm)	7.0
Discharge port diameter (mm)	5.0
Suction valve disc diameter (mm)	10.0
Discharge valve disc diameter (mm)	8.0
Operating Conditions	
Discharge pressure (bar)	10
Suction pressure (bar)	1
Suction temperature (degree Celsius)	50
Discharge temperature (degree Celsius)	127
Coefficient of friction	0.2
Motor operating speed (rpm)	3000

Effects of Number of Cycle per Motor Revolution

Simulations were performed to study the effects of stroke length, h, motor revolution speed, ω and the number of compressor cycles per motor revolution, N. The effective compressor swept volume was kept constant by

(a) maintaining a constant product of $h \times N$, see Table 2, or
(b) maintaining a constant product of $\omega \times N$, see Table 3.

Table 2 shows that by increasing the number of compressor cycle per motor revolution, N, the amount of P-V work and mass per cycle decrease. This is because the stroke length, h becomes shorter. The results also show that the gas pressure torque increases with N, due to an increase in the gas pressure as the result of more valve losses. The increase in the inertia torque is a result of higher piston acceleration due to shorter cycle time. The final overall result is the lower overall throughput.

Table 2: Effects of increasing *N* but reducing *h* to keep a constant $h \times N$.

No. of compressor cycle per motor revolution, N	1	2	3
Stroke length, *h*	18.0	9.0	6.0
Normalised P-V Work per cycle	1.0	0.5167	0.3492
Normalised Volumetric Efficiency	1.0	1.0121	0.9993
Normalised Mass induced per cycle	1.0	0.5060	0.3331
Normalised Mass delivery	1.0	1.0121	0.9993
Normalised Inertia torque	1.0	2.000	3.000
Normalised Gas pressure torque	1.0	1.0124	1.0376
Normalised Overall Torque	1.0	1.1675	1.3376
Normalised Overall Power	1.0	1.1674	1.3376
Normalised Overall Throughput	1.0	0.8669	0.7471

Table 3 shows the case for increasing *N* but reducing compressor speed. The P-V work and mass induced per cycle are approximately constant when *N* increases. Both of them may have a slight increasing trend with N as observed in Table 3. Table 3 shows that reducing the motor speed ω while N increases can result in better compressor performance.

Table 3: Effects of increasing *N* and reducing ω to maintain a constant effective swept volume.

No. of compressor cycle per motor revolution, N	1	2	3
Motor revolution speed (rev/min)	3000	1500	1000
Normalised P-V Work	1.0	1.0050	1.0063
Normalised Volumetric Efficiency	1.0	1.0059	1.0084
Normalised Mass induced per cycle	1.0	1.013	1.0154
Normalised Mass delivery	1.0	1.0059	1.0084
Normalised Inertia torque	1.0	2.0000	3.0013
Normalised Gas pressure torque	1.0	1.990	2.9819
Normalised Overall torque	1.0	1.7028	3.2726
Normalised Overall Power	1.0	1.000	1.0908
Normalised Overall throughput	1.0	1.0051	0.9244

Effects of Profile Pitch Diameter

Profile pitch diameter, D_f is equal inner piston diameter plus the depth of the groove of the motion profile. Figure 5 shows that the overall torque increases with decreasing D_f. A decrease in D_f increases the pressure angle and hence increases the frictional force F_f, as shown in Figure 6. Hence the piston with bigger inner piston diameter is preferred.

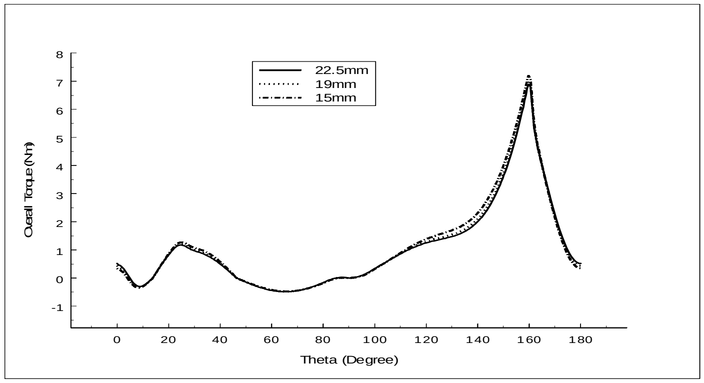


Figure 5: Effects of profile pitch diameter on overall torque

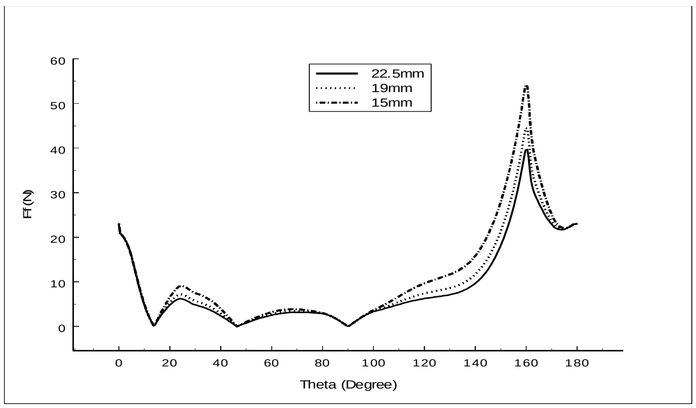


Figure 6: Variation of friction force F_f

Effects of Piston Diameter and Stroke Length

Table 4 shows the effect of varying the piston diameter and stroke length simultaneously such that the swept volume of the compressor remains constant. The difference between the piston diameter and the profile pitch diameter is fixed at approximately 3mm. In addition, the clearance volume is set at one percent of the swept volume.

Table 4: Effects of piston diameter and stroke length

Stroke length	18	21	24
Piston Diameter, mm	25.4	23.52	22.00
Profile pitch diameter, mm	22.5	20.62	19.1
Normalised P-V Work	1.0	1.0057	1.0010
Normalised Volumetric Efficiency	1.0	1.0052	0.9977
Normalised Mass induced per cycle	1.0	1.0052	0.9977
Normalised Mass delivery	1.0	1.0052	0.9977
Normalised Inertia torque	1.0	1.3611	1.7778
Normalised Gas pressure torque	1.0	0.9947	1.0009
Normalised Overall torque	1.0	1.0350	1.0965
Normalised Overall throughput	1.0	0.9712	0.9099

It can be seen clearly that most of the performance parameters remain relatively constant except the inertia torque which increases with the stroke length. The magnitude of the velocity increases with the stroke length, so as the acceleration. The increase in pressure angle and the acceleration caused an increase in the inertia torque and hence the overall torque. Thus, the overall throughput decreases.

ROTAPROCATING COMPRESSOR OPTIMIZATION

In the present study, the objective function is defined as

$$F_{opt} = \frac{\text{Mass per cycle (g)}}{\text{Total work input (J)}} \tag{3}$$

The choice of the above objective function sets the optmisation search to maximise the mass flow and at the same time minimise the work input. In any optmisation study, the selection of free variables and their corresponding limits are very important because they dictate the boundary of the feasible region. Only parameters that give significant effects on the objective function should be chosen. Table 5 shows the free variables with their respective limits set in the study. The lower and upper limits of the piston diameter are set to be approximately 0.7 to 1.5 times of the original piston diameter respectively. The stroke length is selected to keep the capacity of the compressor constant.

The geometrical constraints are shown in Table 6, these parameters ensure that a particular geometrical configuration is physically feasible. In this case, the limits of the valve disc diameters are selected such that they are always greater than the port diameter by at least three millimetres to ensure that the port is well sealed. The total sum of diameters for both ports is set to be less than the diameter of the piston.

Table 5: Free variables and their limits.

No.	Constraint	Constraint Function	Initial Design	Unit
1	Piston Diameter	$18 \le D_P \le 40$	25.4	mm
2	Suction Port Diameter	$3 \le D_{SP} \le 12$	7	mm
3	Discharge Port Diameter	$3 \le D_{DP} \le 10$	5	mm
4	Suction Valve Disc Diameter	$6 \le D_{SD} \le 16$	10	mm
5	Discharge Valve Disc Diameter	$6 \le D_{DD} \le 16$	8	mm

Table 6: Geometrical constraints

No.	Constraint	Constraint Function
1	Suction Valve Disc Radius	$D_{SP} + 3 \le D_{SD} \le D_{SP} + 6$
2	Discharge Valve Disc Radius	$D_{DP} + 3 \le D_{DD} \le D_{DP} + 6$
3	Sum of Port Diameters	$0 \le D_{SP} + D_{DP} \le D_P$

RESULTS AND DISCUSSION

In the optimisation study, the variations of the normalised value (w.r.t. the initial design) of the free variables and performance parameters are shown against the feasible iteration number. The feasible iteration number is the number of successful search where the design satisfies all the constraints imposed. Figure 8 shows that the optimisation process requires 286 executions of the simulation model but only 92 of them lie in the feasible region.

Figure 8 shows that the objection function value increased by approximately 11 %. This is achieved by increasing the mass flow by 12 %, Figure 9, and reducing the power by 9%, Figure 10. The reduction in power input is obviously not caused by the reduction in the PV work (Figure 11) neither it is caused by an increase in the mass flow (Figure 12). As shown in figure 13 that the increase in the volumetric efficiency is only 1.2%.

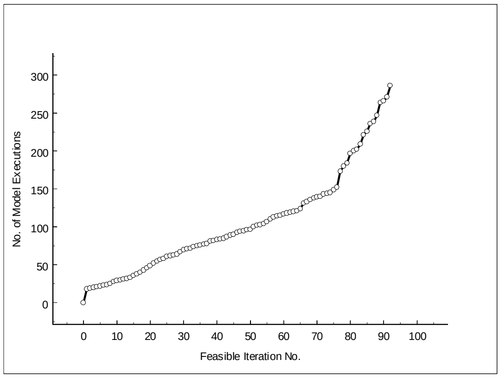


Figure 7: Variation of number of model execution

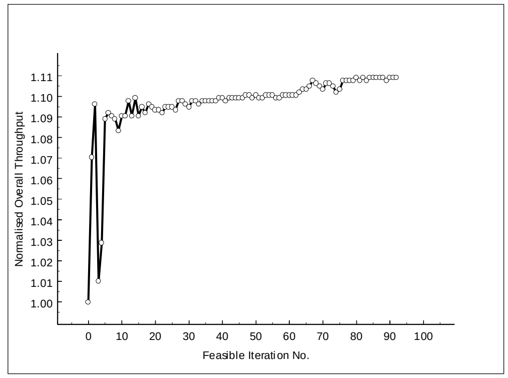


Figure 8: Variation of Normalised Overall Throughput

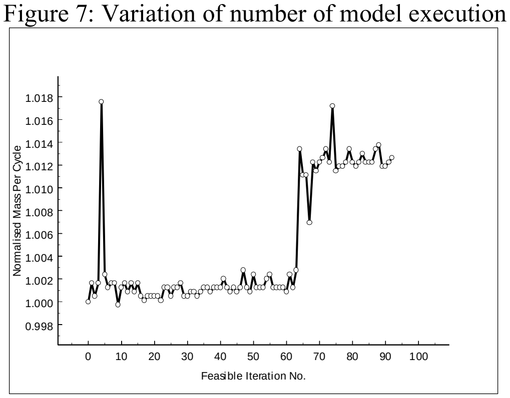


Figure 9: Variation of normalized mass per cycle

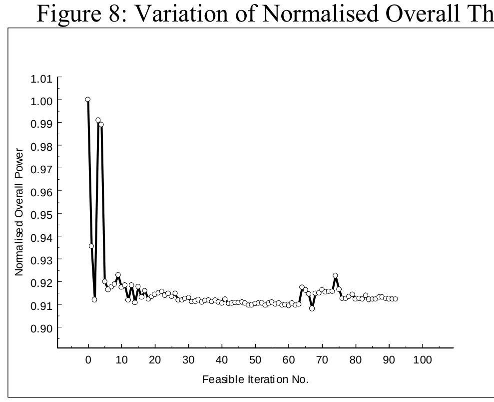


Figure 10: Variation of Normalised Overall Power

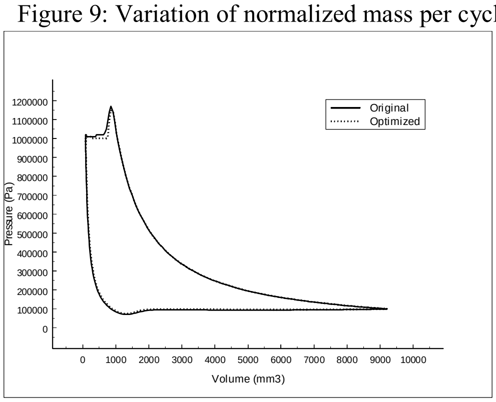


Figure 11: Comparison of PV Work

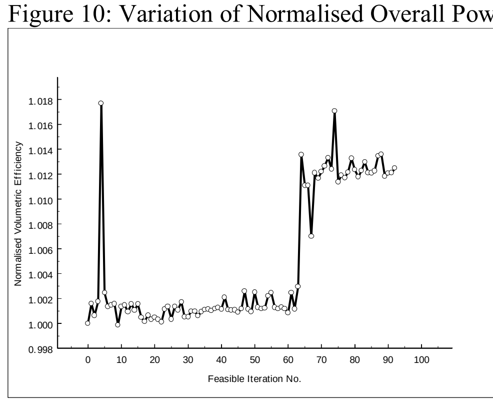


Figure 12: Variation of normalized volumetric efficiency

Figure 13 explains that the decreased in the power is mainly due to a significant reduction in the value of the overall torque. The reduction in overall torque is in turn caused by the reduction in the inertia torque, see Figure 14. This is due to the reduction in the piston acceleration as a result of a shorter stroke length (Figure 15) which corresponds to a larger piston diameter (Figure 16), as the swept volume was kept constant. Figure 17 shows that the maximum pressure angle decreased by 30% and the optimum value is below 45°, which is a sign of a good design. This reduction is due to the combination of variation in the values of the piston diameter and the stroke length.

CONCLUSIONS

Various motion profiles have been compared and based on the kinematics and mechanical performance criteria the modified sinusoidal profile is selected for the present study. Results of the optimization studies suggested that the optimum compressor to have a 20% larger piston diameter and 35% shorter stroke length while keeping the swept volume constant. This has the effect of reducing the overall power by about 9% and resulted in an increase in the overall throughput of 11%. The gain is mainly come from the reduction in the driving torque due to an optimum combination of geometrical dimensions.

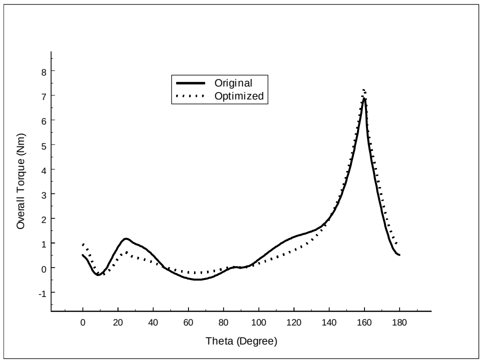


Figure 13: Comparison of overall torque

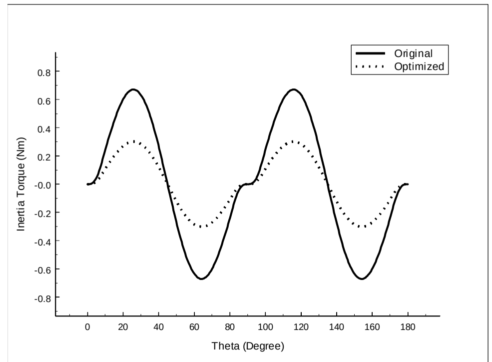


Figure 14: Comparison of inertia torque

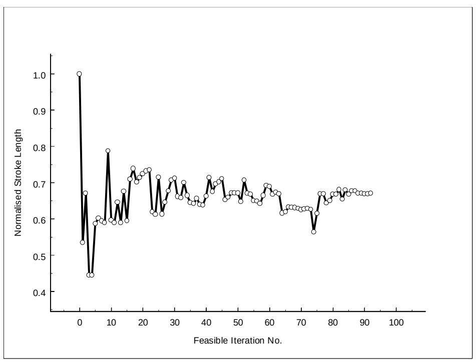


Figure 15: Variation of normalized stroke length

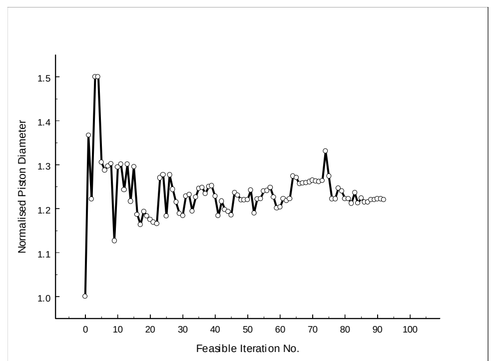


Figure 16: Variation of normalized piston diameter

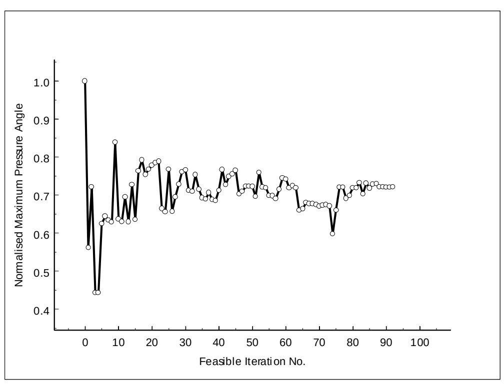


Figure 17: Variation of normalized maximum pressure angle

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