COMPUTER AIDED KINEMATIC SYNTHESIS FOR DESIGN MODIFICATIONS TO A MANUALLY OPERATED RICE TRANSPLANTER

by

SOE TINT

A thesis
presented to the University of Manitoba
in partial fulfillment of the
requirements for the degree of
Master of Science
in
The Department of Agricultural Engineering

Winnipeg, Manitoba

ISBN Ø-315-33922-5

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MASTER OF SCIENCE

RICE GODDESS

O, Rice Goddess, come up to the rice bin.
Do not go astray in the meadows and fields,
for mice to bite you and birds to take you
in their beaks. Go to the happy place to rear
your children and grandchildren in prosperity.
Come!

(Lucien M. Hanks)

THE WORLD FOOD PROBLEM

The world food problem lies in the fact that God ceased to create land long ago but He is still creating people. Even though there are still lands available for possible expansion of world rice areas in the near future, the land nevertheless, is a fixed resource. Conversion of marginal land for rice cultivation furthermore, is costly. Increases in future world rice supply rests, therefore, heavily upon increases in per unit land yield.

(Anthony Baily)

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ABSTRACT

World production of rice is lagging behind global population growth in general. To fill this gap, newly developed high yielding varieties (HYV) of rice were introduced in many countries. However, the high yielding varieties require more labor and more associated technologies. Machine transplanting can help overcome some of the additional requirements.

The International Rice Research Institute developed a manually operated transplanting machine that can be fabricated in local workshops. At the same time the machine complements the HYV. The machine field capacity and performance have been acceptable. However, since there are no perfect machines, improvements with advancing technology are always possible. In the existing machine the reciprocating movement of the seedling tray was achieved by passing a roller chain, connected by pin and a short link to the seedling tray, around a pair of end rollers. For each stroke of the drive lever, a corresponding linear tray movement was achieved. However, equal linear movement was obtained only while the pin joint was between the centres of the rollers. The results were that nonuniform seedling numbers and sometimes empty hills were produced. The percentage of missed hills was estimated at about 5 percent.

The modifications to the indexing mechanism resulted in a four bar linkage, cam and trip mechanism. The trip mechanism which changes the direction of tray travel was activated by the action of a plate cam and an end face cam. The required angular movement of a cam shaft was sup-

plied from the handle assembly through a four bar linkage. The four bar linkage was constructed to provide for maximum output angle at optimum transmission angle for both a flat-faced and a roller follower. The size of the cam was designed to suit both types of follower. Minimum possible cam size was chosen with optimal offsetting.

Cycloidal motion with finite jerk and a continuous acceleration curve was selected for the follower motion. For the trip mechanism the overhang length and guide length were specified on the basis of setting the maximum working pressure angle at one half of the maximum allowable pressure angle. All design calculations were done through the application of computer programs.

The new design performed well in limited testing. No unequal linear movements nor lost motion can be observed. Since the whole indexing assembly was replaced, lack of rigidity at the connecting pin joint was also eliminated. By eliminating potentially missed hills this design improved performance by a minimum of 5 percent. The overall improvement due to greater uniformity of plant population and saved labor for filling missed hills is even more significant.

ACKNOWLEDGEMENTS

The author would like to express his sincere appreciation to his adviser, Dr. J. S. Townsend, for his encouragement, suggestions and help given during the course of studies and preparation of this thesis. Even though he was involved with other work he made the necessary corrections with patience. Thanks are extended to Professor L. C. Buchanan for his helpful comments. Thanks are especially due to Dr. W. L. Cleghorn for his sincere help, guidance and keen interest in the subject. The author is very grateful to all his teachers from monastary to university. Suggestions, discussions and technical assistance from Mr. J. Putnam, Mr. B. Morgan, and Mr. H. Kroker were invaluable. Their help in this work is gratefully acknowledged. Thanks go to all his fellow students who spared their valuable time and helped in numerous ways and created pleasant working conditions. It is impossible to convey adequately the recognition owing to all concerned. Among the many are Q. Lei, Chris Hawthorne and Tom Naylor. The author wishes to express sincere gratitude to Mrs. Y. Townsend, Mrs. A. Skene, Mrs. H. Rhoda and their families for their constant help throughout his stay in Winnipeg.

The author also acknowledges the support of the Government of the Socialist Republic of the Union of Burma and The Canadian International Development Agency through the International Rice Research Institute for graduate studies and the support of his parents for undergraduate studies.

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Chapter I

INTRODUCTION

1.1 THE ROLE OF RICE

Cereals are the main staple food for humans. The most common cereals are rice, wheat, maize(corn), and millet. Rice is the main staple for about 67 percent of the world's population. The energy content of rice is 14.9 MJ/kg. This energy content is much higher than other cereals. The protein content is 7.5 percent by weight which is less than wheat. But rice compensates for the lower protein content by its very high yield. Based on 1982 average yields for the world production of rice, one hectare of rice produces approximately 30,800 MJ of energy, 35 kg of fat, 155 kg of protein, and 1606 kg of carbohydrates. These amounts are higher than any other staple cereal food crop. Table 1 summarizes comparisons of rice to other cereals.

Rice accounts for up to one half of the daily energy intake in many Asian countries. It is also a major source of protein for most Asian people. Of all cereals, rice is the basic food for most people closely followed by wheat. Rice is important not only as a staple food but as the basis of agricultural production and trade. Approximately 90 percent of the world's total rice production is eaten by the people who produce it (Luh, 1980). Less than 5 percent of the world's annual supply of milled rice moves in international trade (Chandler, 1979).

TABLE 1

Nutrient value of rice compared to other cereals

Comparison unit	Rice (white milled)[1]	Wheat (white flour)[2]	Maize (corn meal)[3]	Millet (grain)
Yield kg/ha	2067	1467	1954	681
MJ/kg	14.9	14.8	14.8	14
MJ/ha	30,800	21,660	29,030	9,057
Fats g/kg	17	10	11	15
Fats kg/ha	35	15	21	10
Carbohydrate g/kg	777	747	788	780
Carbohydrate g/ha	1,606	1,096	1,540	531
Protein g/kg	75	112	75	56
Protein kg/ha	155	164	147	38

Source: FAO (1982), Source: Luh, (1980)
1 converted from paddy (rough rice) at 72 percent
2 converted from whole grain at 73 percent
3 converted from shelled corn at 56.4 percent

1.2 POPULATION AND RICE PRODUCTION

In 1982, the population of the world was 4.59 billion and was increasing at a rate of 1.7 percent per year. Rice production was 41.2 billion metric tonnes but was increasing by only 0.044 percent per year (FAO, 1982). The population of the Asian countries was 2.67 billion in 1982 and was increasing at a rate of 1.78 percent while rice production was decreasing at a rate of 0.183 percent. Population growth affects the type of crops grown. Due to dense population, sweet potatoes have replaced rice in some areas of Java and cassava has replaced traditional yams in parts of Africa.

The world's almost static output of rice is lagging increasingly behind the needs of an expanding global population. This is undesirable. If the situation continues, starvation or malnutrition might be expected in rice consuming Asian countries. The Food and Agricultural Organization of the United Nations (FAO) has estimated that it will be necessary to increase the production of rice by 3 percent annually until the year 2000 to keep pace with the demand of an ever increasing population (UNESCO Journal, 1984).

1.3 SOLUTIONS TO THE PROBLEM OF RICE SUPPLY

One approach to increasing the rice supply would be to increase the rice growing area, either extensively or intensively. Due to annual increases in population and other related factors, the rice growing area of the world has been reduced year by year. A study conducted by FAO indicated that the world's total rice growing area was reduced by 1.624 million hectares in 1982. Almost all of this reduction occurred in Asian countries. It appears that the extensive method for increasing production of rice is not realizable.

The intensive method of increasing rice production is to promote multiple cropping systems over one cropping year. The term multiple cropping simply means a sequence of crops grown on the same plot. At a particular stage in the growth of the first crop, perhaps at harvest time or a week before or after, a second crop is planted. As the second crop reaches a certain point in its maturation the third crop is planted and so on.

If cereal crops are grown continuously on the same piece of land without a break and without the use of fertilizers or manure crop yields will decline as the initial soil fertility is exhausted. Plant diseases specific to the crop become endemic and it is sometimes difficult to control weeds. A period of fallow, where the land has no crop for a season or more, has been incorporated in many traditional systems.

Modern agronomy has disproved the idea that soil has to be fallow between crops. In modern agriculture, using new technologies, continuous cereal cultivation is possible. Since 1962 the International Rice Research Institute (IRRI) has harvested 25 crops of rice from the same piece of land and annual yields have been maintained at about 21 t/year (Grigg, 1980). This represents the intensive use of land.

Where temperature and other factors favor continuous year-round growth, the intensive method could be the solution for Asian countries. Johnson (1966) reported that in many areas three crops can be harvested yearly instead of a single crop produced by hand labor. To fully utilize the intensive method, appropriate mechanization will be necessary depending on the type of crops in the cropping system and the number of crops per year.

Another way to increase production would be to increase yield per unit area. In 1982, the average yield in Asian countries was 2923 kg/ha, just above the world average of 2871 kg/ha (FAO, 1982). The average yield in developed countries was above 5000 kg/ha. This comparison indicates the potential yield for those countries that now produce at low average yields.

Considerable expansion in mechanization and improved cropping systems could lead to increased production of rice to meet the need of an increasing population. Modern technology to promote this trend involves the development of new grain varieties, the use of fertilizer, the adaption of irrigation and drainage practices, the use of chemical weed control methods, the practice of increased cropping intensity, the use of appropriate mechanization, and the minimization of post harvest losses in storage and processing (Lapp, 1977). These interrelated factors of the intensive method, if properly applied, should result in higher overall production.

1.4 THE ROLE OF MECHANIZATION

Newly developed rice varieties have special peculiarities. The plant growing season is short and the plant height is short. Better yields are possible with proper management of water, application of fertilizer, and good weed control. With increased cropping intensity timeliness is another factor contributing to better yield. To achieve timeliness it may be necessary to use appropriate mechanization to replace indigenous cultural practices that limit the productivity of a given land base and water resource. To achieve the promise of the new varieties machine transplanting should be considered a first priority in the adoption of appropriate mechanization.

Mechanization can help achieve increases in yield. But mechanization alone cannot sustain increased yields. All generalizations about the usefulness of mechanization must be modified to fit local circumstances. Some technologies transferred from developed countries have not been useful in developing countries. Sometimes the technology was designed to maximize labor productivity but labor productivity may not be crucial in the developing countries.

In some developing countries, much has been accomplished by importing machines and equipment from developed countries. Often these imports are not suited to small farm conditions. Major manufacturers have little incentive to design equipment specifically for these developing countries because the size of the market in any one country is relatively small.

Advanced, sophisticated machinery, although adequately designed for general use, would not be acceptable where the mechanical aptitudes of the local operators are very low. Most farmers in developing Asian countries are interested only in cheap, simple power equipment that they can afford. The diminishing supply of liquid petroleum fuels and the increased cost of fuel has stimulated more interest in manually operated machines. Therefore, the manually operated transplanting machine becomes the focal point for agricultural engineers in promoting the mechanization of rice transplanting in developing countries.

1.5 OBJECTIVE OF THIS STUDY

Some rice growing countries in Asia have tested and modified a manually operated transplanter designed at IRRI (Model TR1). Some users have insisted on the use of traditional washed-root seedlings while some have tried to avoid the use of the wooden growing frames in seedling mat production. Research on favorable conditions of soil, water depth, plant size and seedling age has been done. But no one has tried to eliminate the inherent dead spots in the indexing mechanism on the seedling tray drive.

This problem has been a concern in every country using the machine. The dead spots, where the picker punches through the same position on the seedling mat twice, results in nonuniform seedling numbers per hill and in many cases produces empty hills. If a repeated stroke is used the resulting hills may have more seedlings than required.

The objective of this study is to redesign the indexing mechanism for the tray drive in order to overcome the problem of missed hills and nonuniform seedling numbers per hill. Elimination of the inherent indexing problem can improve hill setting efficiency by as much as 9 percent. This improvement does not appear to be significant but in terms of rice supply the result would be enough milled rice per hectare to feed 2 to 3 people for one year on the average in rice growing countries.

Chapter II

LITERATURE REVIEW

2.1 BACKGROUND

It is unclear exactly where and when rice first evolved. In India the earliest traces of rice date no earlier than the first millenium BC (Hanks, 1972). Nevertheless, both botanists and archaeologists agree that rice must have grown wild since prehistoric ages in East Asia. The wild-growing rice was probably a supplemental food for prehistoric people. Since rice has been known to man, four techniques have been used to culture it. These techniques are gathering, shifting cultivation, broadcasting, and transplanting. These methods of producing rice were used consecutively and/or simultaneously.

The earliest utilization of rice was gathering followed by shifting cultivation. For gathering people obtained food for themselves from the scattered and unpredictable stands of rice and other plants. Hunting and fishing were also part of gathering. The insufficient and irregular supply of foods seems to have led to agriculture and consequently the gathering stage changed to shifting cultivation. Rice culture eventually evolved to broadcasting and transplanting. Transplanting was probably used in China about the time of the birth of Christ (Luh, 1980). Archaeologists have found traces of diked rice fields near Pagan, an ancient capital city of Burma, in Southeast Asia. The city dates back to 1000

AD. In the Red River delta of Vietnam transplanting seems to date that far back also.

2.2 TYPES AND NATURE OF RICE

Rice belongs to the extensive family of grasses or graminae. Rice can be classified into two types- Indica and Japonica. The Indica types have long-grain kernels, tend to be nonphotoperiod sensitive and are responsive to temperature whereas Japonica varieties have oval-shaped kernels, are photoperiod sensitive and continue to grow inspite of fluctuations in temperature. Rice is widely dispersed around the world in latitudes from 50 degrees north to 40 degrees south and at altitudes from sea level to more than 2,500 meters.

Rice evolved in the tropics as a semi-aquatic plant. Depending on the variety, rice matures in from 90 days to more than 300 days. The traditional rice varieties of the tropics were excessively tall with very long drooping leaves. The height usually ranged from 1600 to 2000 mm. When fertilization and other modern management practices were used with traditional rice varieties, the rice tended to lodge before harvest. Earlier research had established a direct negative correlation between grain yield and number of days before harvest that lodging occured. That is, the earlier the lodging, the lower the yield. Therefore, the approximate average yield of traditional varieties has been limited to about 1500 kg/ha.

2.3 HIGH YIELDING VARIETIES AND THEIR DEVELOPMENT

Professor Malthus, in his essay on the Principle of Population published in 1789, stated,

> "I say, that the power of the population is indefinitely greater than the power in the earth to produce subsistence for man.
>
> Population, when unchecked, increases in a geometrical ratio. Subsistence increases in an arithmetical ratio."

His statement appears to be becomming more evident during the 20th century. Within three decates the population increased more than 66 percent.

Jay Richter, an Associate Director of Agriculture Services, wrote an article on "Our Lazy Acres Can Yield Far More Food" in the New York Times on October 9, 1949. He commented on Malthus roughly as follows. Malthus envisaged a world of constant misery that could not produce enough food to match the rate of population growth. Now the production of enough food is possible through widespread application of modern technology. Malthusian theory might be still valid, however, it is more questionable.

Mellor (1962) points out that urban population is increasing in many developing countries faster than the available food supply. More specifically, population increases in most rice consuming countries are among the highest in the world and consequently demands for rice are expected to increase by 30 percent by 1987 (Luh, 1980).

Yearly rice consumption in many rice growing countries averages 94.7 kg/person compared to a world average of 58.4 kg/person. Statistical comparisons have stimulated agronomists and plant breeders to produce better yielding varieties of rice. Because of the dedication of rice breeders a new era of agriculture was possible in the early 1960's with the release of new High Yielding Varieties (HYV) of rice. Dr. Chandler who was head of IRRI in the Philippines where the HYV's were developed commented:

"Only three years ago people were screaming, 'How are
we going to feed the teeming millions people of Asia ?; now some
of those people are yelling about over production" (Shabecoff, 1970).

The HYV rices were so successful that the total world production of rice increased more than 50 percent between 1960 and 1975. Note that the increase was due not only to high yielding varieties but also to other associated technologies as well. The success in rice was primarily based on similar achievements with high yielding varieties of wheat which were released after World War II (Chandler, 1973).

The newly developed rice varieties were short in height (approximately 800 to 1200 mm) and were stiff-strawed. These were improvements over the traditional varieties to reduce lodging and to increase the grain/straw ratio. It has been recognized that nothing has contributed more to the increased yield potential of rice than the drastic changing of plant architecture. The significant decrease in plant height was accompanied by a major increase in yield potential caused primarily by the more efficient use of solar energy and soil nutrients.

Another advantages of HYV rice is that there is a high percentage of fully filled kernels on each panicle. The percentage of filled grain tends to remain constant even when extra fertilizer gives larger panicles. Traditional rice does not respond the same and often the percentage of filled grain will decrease with increased fertilization. With HYV rice the losses due to lodging or over-luxuriant growth rarely occur and narrow row spacing is possible. Matsushima (1980) suggested that the use of HYV rice and narrow row spacing contribute to higher yield.

HYV rice has other desirable characteristics. It is highly fertilizer responsive and is relatively insensitive to photoperiod. Lack of photoperiod sensitivity allows a variety to be planted over a wide latitude with little variation in growth duration. In brief HYV rice grows rapidly and starts forming tillers sooner resulting in a maturity period of about 100 - 125 days from seed to seed without sacrificing grain yield.

2.4 PLANTING METHODS AND THE EFFECT ON YIELD

In Asia, generally both broadcasting and transplanting are commonly practiced. The broadcasting method requires less labor and time whereas the transplanting is complex and time consuming. In Japan 220 man-h/ha of labor input are needed for hand transplanting (Ezaki, 1963). Other researchers have reported that hand transplanting requires 250-300 man-h/ha or roughly 25 percent of the total labor requirement for the crop (RNAM, 1979). Kim (1977) reported that hand transplanting has a higher input energy requirement than that required for broadcasting, as much as 26 times as much.

Although the input energy requirements for the two planting methods are different several researchers have reported that yields from both methods are comparable and are almost equal provided proper management is provided (Jayasekara, 1966). In terms of the ratio of output of rice to input of work more rice is produced by broadcasting than transplanting. Broadcasting appears to be 1.65 times more efficient than transplanting. However, on the average, the output from the transplanted rice exceeds the output from the broadcast rice (Hanks, 1972).

Broadcasting is cheaper than transplanting in terms of both labor and cost. It has the disadvantage of nonuniform plant spacing resulting in nonuniform plant population density which could lead to uneven maturity. Broadcasting requires higher management for good weed control. It has been reported that lodging of the broadcast crop occurs due to poor root development (Jesus De, 1959). Singh (1974) found that rice plants sown by broadcasting were susceptible to lodging with only slightly adverse enviromental conditions. Transplanting saves time of residence of the rice in the field faciliating multiple cropping systems. Uniform plant populations and better water and weed management can be achieved through transplanting. Less seed is required. Conservation of nitrogen is possible with this method.

Transplanting becomes more important as population increases and land becomes scarce. The farmer must select the correct method to use based the situation he faces. He must balance the advantages and disadvantages of each system.

If water and labor are critical then broadcasting is usually indicated. If multiple cropping is desired transplanting is preferred. These are not the only considerations because there are other secondary factors to consider. The above discussion is especially pertinent with traditional varieties.

With HYV rice an additional reason for transplanting is to ensure a sufficient number of plants for tillering in the early growth period. This is because high yield depends on the number of grains and the number of grains depends on the number of productive tillers. Greater numbers of tillers can be achieved through transplanting. Broadcasting is not a desirable practice for promoting high yield in HYV rice. Yield can be expressed as a function of the following parameters:

$$\text{Yield} = (\text{grains/panicle})(\text{panicles/ha})(K) \qquad [2.4\text{-}1]$$

where,

$$K = \text{matured grains/total grains}$$

Obviously, the number of grains, the number of panicles and the percentage of ripened grain greatly influence final yield. The number of panicles depends directly on the number of tillers produced in the early growth period. Panicle bearing tillers contribute directly to yield. Generally, HYV rice has less tillering capacity than traditional varieties (Matsushima, 1976).

Tillering depends on the average soil temperature and the difference in the soil temperature from day to night. High average soil temperature helps the rice roots accelerate the absorption of nutrients for growth.

Since high soil temperature and high day-night soil temperature differences are most pronounced at shallow soil depths, shallow planting is desirable.

2.5 ADVANTAGES OF MACHINE TRANSPLANTING

Hand transplanting is a back breaking job. It requires 120 man-h/ha in traditional rice cultivation (Johnson, 1962). HYV cultivation will require even more labor because narrower row spacing and straight row planting is desirable. The labor requirement is usually maximum at transplanting season. Machine transplanting will minimize the labor requirement and result in timeliness.

Machine transplanting is even more desirable in HYV cultivation. Some associated technologies of HYV culture are compatible only with machine planting. For example, less root damage and better weed control are possible with machine planting.

Timeliness is another factor in obtaining better yield. Shallow transplanting and narrow row spacing, which contribute to better yield, can be achieved through machine transplanting. This can be seen clearly from the report of the Extension Division, Agriculture Corporation, Burma. The report (Townsend, 1982) explains that planting densities of over 300,000 hills per hectare were achieved with machine planting. The average yield for the machine transplanted area was 4.6 t/ha compared to the national average of 2.7 t/ha. Therefore in many Asian countries there is an increasing interest in mechanical transplanting (Singh, 1973). However, the use of machines is limited by many factors in developing countries.

The first factor is the initial cost which farmers may not be able to afford. Secondly, many small farmers cannot use such a machine enough hours per year to justify the purchase. Custom operators may not be able to use the machine efficiently. Labor productivity is not crucial in many developing countries. The diminishing supply of fossil fuels is another consideration. Research on the use of powered transplanters, conducted by the International Rice Research Institute (IRRI), found that powered transplanters may be too costly for rice farmers. The high capacity of the transplanters may result in labor displacement (IRRI, 1977).

But farmers want machines in order to reduce labor requirements. They also want machines with low energy consumption. Therefore, the use of a small manually operated machine is the logical choice. Small machines are also indicated where the average individual field size is about 600 square meters (Johnson, 1965). Manually operated rice transplanters have been in use in many rice growing Asian countries. These machines were intended to solve the problem of labor shortages at transplanting time. The use of machine transplanting becomes ever more important for the newly developed HYV rices with their special requirements as noted above.

The earliest developments in transplanting machines were recorded in Japan in 1898 (Shin-Norinsha, 1971). China was second in developing manually operated rice transplanters over the years 1950 - 1960. Development of machines in India and England came later (RNAM, 1979). The working capacity of the first Chinese transplanters was about 0.3 ha/day in well prepared and levelled fields using 3 persons (RNAM, 1979). In

1962, in Tamilnadu Agricultural University in India, a 2-row manual rice transplanter was developed. However, it failed to work satisfactorily (RNAM, 1979). In many developing countries lack of suitable agricultural machinery designs that can be produced by simple fabrication methods is a dominant problem (Kham, 1970).

2.6 TRANSPLANTER DEVELOPMENT AT IRRI

The Agricultural Engineering Department at the International Rice Research Institute in the Philippines began development of a manual rice transplanter in 1977. The design was intended to be fabricated by local workshops. The designed was based on the Chinese developed transplanter. The prototype was released to many rice growing countries. IRRI also offered farmers and manufacturers free technical imformation and assistance in the use and manufacture of the machine. IRRI encouraged manufacturers and dealers to incorporate modification to the basic design if necessary.

After noting the experiences with the machine in different countries over several years, IRRI has improved some components of the machine. The recently improved transplanter (Model TR1) was produced in 1981. The IRRI rice transplanter consists of a wooden skid, a main assembly that supports the seedling tray, an indexing assembly, a pivoted picker bar assembly and a handle. This machine is manually operated and is pulled through the field by the operator. An overall view of the transplanter is illustrated in Figure 1. Figure 2 shows an exploded view of the Model TR1 with a list of all the parts. The transplanter's specification are listed in Table 2.

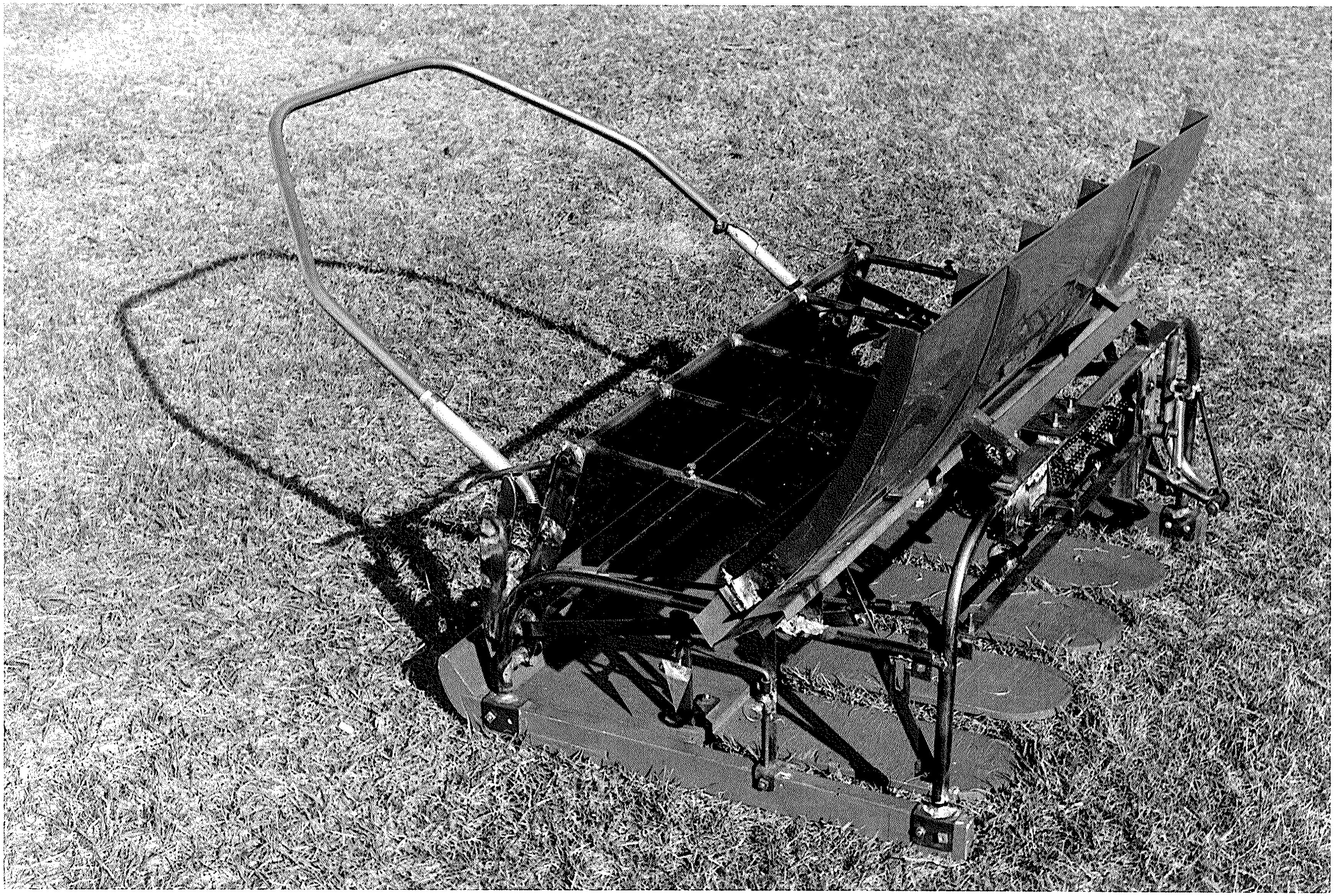
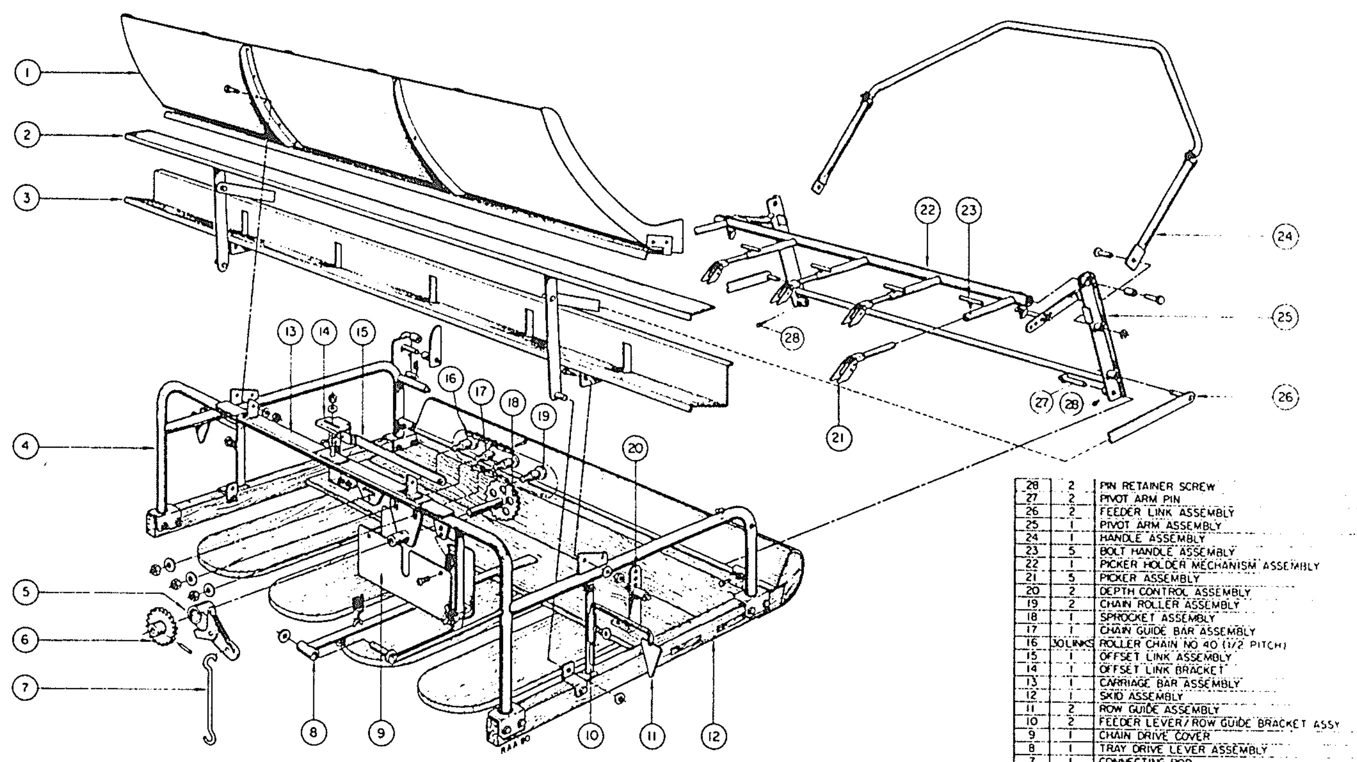


Figure 1: The rice planter TR1



Exploded view of rice transplanter.

REF.	REQ'D	DESCRIPTION
28	2	PIN RETAINER SCREW
27	2	PIVOT ARM PIN
26	2	FEEDER LINK ASSEMBLY
25	1	PIVOT ARM ASSEMBLY
24	1	HANDLE ASSEMBLY
23	5	BOLT HANDLE ASSEMBLY
22	1	PICKER HOLDER MECHANISM ASSEMBLY
21	5	PICKER ASSEMBLY
20	2	DEPTH CONTROL ASSEMBLY
19	2	CHAIN ROLLER ASSEMBLY
18	1	SPROCKET ASSEMBLY
17	1	CHAIN GUIDE BAR ASSEMBLY
16	30 LINKS	ROLLER CHAIN NO 40 (1/2 PITCH)
15	1	OFFSET LINK ASSEMBLY
14	1	OFFSET LINK BRACKET
13	1	CARRIAGE BAR ASSEMBLY
12	1	SKID ASSEMBLY
11	2	ROW GUIDE ASSEMBLY
10	2	FEEDER LEVER/ROW GUIDE BRACKET ASSY
9	1	CHAIN DRIVE COVER
8	1	TRAY DRIVE LEVER ASSEMBLY
7	1	CONNECTING ROD
6	1	RATCHET WHEEL
5	1	PAWL LEVER ASSEMBLY
4	1	MAIN FRAME ASSEMBLY
3	1	FEEDING FRAME ASSEMBLY
2	1	SEEDLING FEEDER ASSEMBLY
1	1	TRAY ASSEMBLY

COMPONENT PARTS LIST

Figure 2: Exploded view of Model TR1 rice transplanter
(operator's manual IRRI)

TABLE 2

Specifications of the IRRI Model TR1 transplanter

Model	TR1
Power	One person
Field capacity	0.25 ha per 8-h day
Labor requirement	30 - 35 man-h/ha
Number of rows and spacing	5 at 200 mm
Planting depth adjustment	20 to 50 mm
Field water depth	10 to 100 mm
Seedling age	15 to 30 days
Average missing hills	2 percent
Average seedlings per hill	4 to 6
Mass	25 kg
Length	1200 mm
Width	1160 mm
Construction	Steel and wood

(Source: Operator's manual, IRRI rice transplanter, 1980)

2.7 OPERATION

As the handle is pushed down the pickers move toward the seedling tray. The picker tips go through the feeding frame slots at a preset depth of penetration and each picker punches out a small cluster of seedlings. The picker continues on a downward path and enters the puddled soil, to a depth of 20 to 50 mm. The handle is then pulled back toward the operator to remove the pickers from the soil leaving the seedlings inclined at an angle of about 75 degrees with the horizontal. At the end of the stroke, the operator continues to pull the handle to move the whole machine a predetermined distance to the next planting position.

The cycle is repeated to plant another five hills. Each time the handle is pushed downward, the pawl lever indexing assembly moves the seedling tray laterally to ensure that the seedlings are in position to be picked by the pickers during the next cycle. Markers at the side of the machine serve as guides to the operator to ensure straight row planting and uniform spacing within the row. The planting depth and the number of seedlings per hill can be varied by adjustments on the transplanter.

2.8 PROBLEMS WITH THE INDEXING MECHANISM

The maximum lateral travel of the seedling tray is equal to the width of one seedling compartment in the tray. Each compartment is 200 mm wide. A connecting link connects the roller chain to the tray. The seedling tray assembly is mounted on a slide and the connecting link slides the assembly back and forth on the slide. When the operator pushes the handle down, the tray drive lever is activated by the link connected to the picker arm turning the drive sprocket through an angle. For each stroke of the driver, the sprocket has an angular movement and there is corresponding lateral linear movement of the chain. Since the chain is connected to the tray assembly by a connecting link, the tray indexes to the next position (see Figure 3).

However, when the connecting link is at either of the extreme lateral positions of the chain, the lateral movement of the tray is decreased as the connecting link pin moves around the end rollers. This produces nonuniform tray indexing and on some strokes the position of the tray is unchanged. The picker punches through the same position twice resulting in some empty hills. This action could happen twice per tray cycle. Other problems are lack of rigidity and backlash in the indexing linkage, especially at the cotter pin joint.

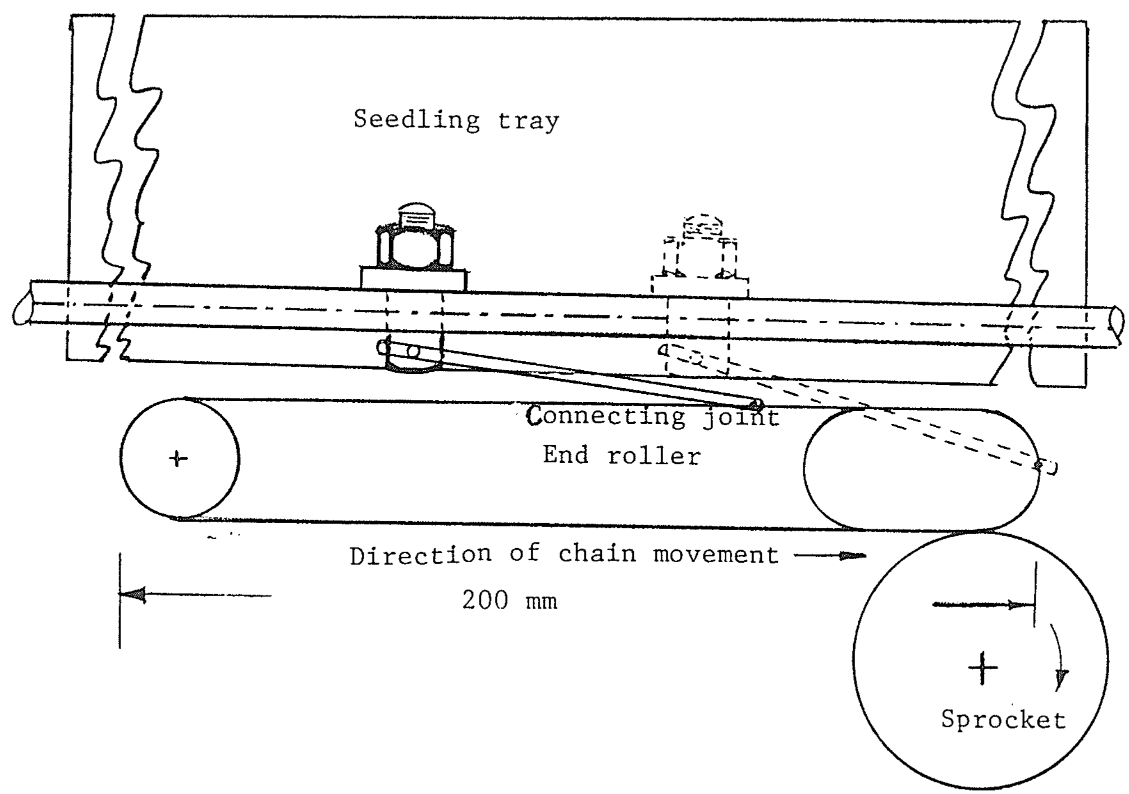


Figure 3: Line drawing of original indexing mechanism

Chapter III

DEVELOPMENT OF DESIGN MODIFICATIONS

3.1 ANALYSIS OF THE ORIGINAL INDEXING MECHANISM

As described above and as illustrated in Figure 3, the seedling tray was connected to the drive chain by a link and a connecting pin. Reciprocating motion of the seedling tray was achieved as the chain progressed around the rollers. Generally the tray travel per stroke was adjusted to be equal to the width of the picker opening. Tray travel that is too large per stroke will leave an unpicked area on the seedling mat while tray travel that is too small per stroke will create an overlapped portion. Therefore, it is desirable to adjust the mechanism as indicated above.

For each stroke of the actuating lever the corresponding lateral movement was observed. The proper movement was obtained only when the connecting pin was between the span of the two end rollers. Unequal lateral movements of the tray were observed when the connecting joint was outside the span of the rollers. The number of the strokes required to complete that portion of the cycle depends on the tray travel adjustment and the size of the end roller. The tray travel is normally adjusted and fixed to the width of the picker opening. Then the tray travel depends only on the size of the end roller when the connecting pin is going around the end roller.

An analysis of the connecting pin travel around the end rollers indicated that smaller rollers would require fewer numbers of strokes resulting in fewer missed hills. A larger roller size would produce more missed hills. Figure 4 shows the theoretical percentage of misses expected for various sizes of pitch diameter of the end rollers (15, 20, 25, 30, 35, 40 mm). In calculating the percentage of misses the miss criteria were based on 2.75 mm, 3.75 mm, 4.75 mm for maximum net lateral tray movement per stroke. From the graph it can be seen that a maximum of 9 percent misses was predicted for a 40 mm pitch roller diameter (4.75 mm criterion).

The desired lateral movement of the tray was always set equal to the width of the picker opening. Since the picker opening is 9.5 mm, a full cycle of tray movement required 41 strokes. In the existing machine, depending on the size of the end roller, a larger number of strokes were observed. A small diameter roller results in fewer unequal lateral tray movements. However the chain link length limits the minimum size of the end roller. On the other hand smooth operation of the chain on the roller can be achieved only with a larger diameter roller.

If it can be assumed that at least three of the chain rollers must be in contact with the end roller when a chain roller is in the extreme position, then the minimum diameter for the end roller can be defined as:

$$Dm = 1.414\ P \tag{3.1-1}$$

where,

Dm = minimum diameter of end roller, mm

P = the pitch of the roller chain, mm/link

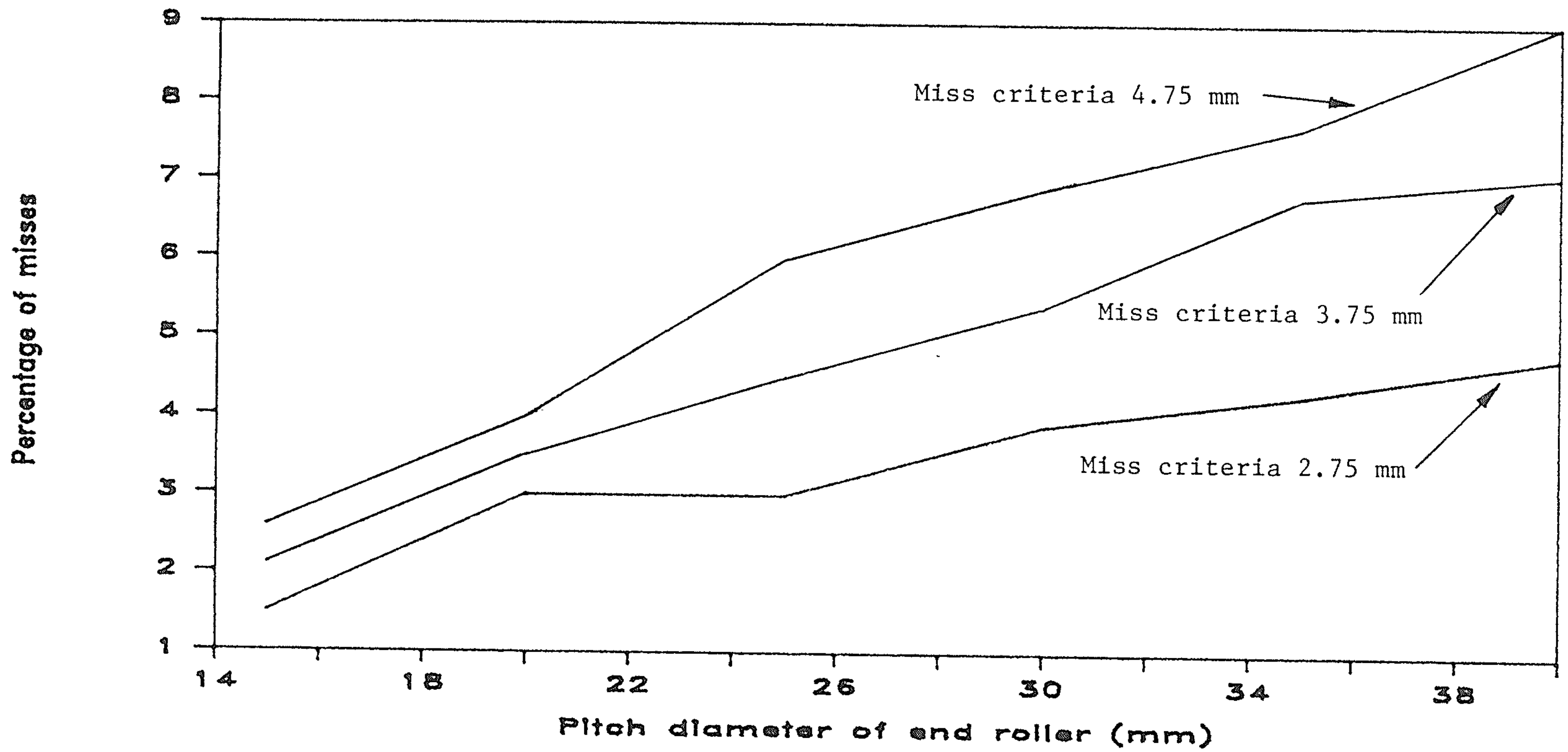


Figure 4: Effect of end roller pitch diameter on percentage of misses

In the original design the lateral movement of the tray per stroke will be equal to the advance of the indexing chain except when the connecting pin is moving around the end roller (two times per cycle). While the chain link with the connecting pin is moving around the end roller the net lateral movement of tray will not be equal to the desired index length. The number of tray movements that will be different than the desired index length will be proportional to the end roller diameter and the desired index length. The different approach positions of the connecting pin will also have an effect on the net lateral tray movement per stroke.

The overall net effect of the variable length of lateral tray movement was to produce hills of varying numbers of seedlings. A computer program was written to illustrate the above phenomenon (see Appendix C). The desired number of seedlings per hill was assumed to be adjustable to 3,4,5 or 6 seedlings. Table 3 shows the predicted number of seedlings per hill compared to the desired number for one set of conditions.

The analysis of Table 3 helps to explain why the observed percent of missing hills has been so variable. Other additional factors such as soil preparation and type, seedling population and machine adjustments all contribute to percent of missed hills. With the new design (see Figure 5), provided that the seedling mat and other machine physical properties remain constant, nonuniform seedling per hill should rarely be produced. Nonuniformity in the number of seedlings per hill will be proportional to the net lateral tray movement as it varies from zero to the desired index length provided that the seedling mats are uniform.

TABLE 3

Predicted number of seedlings per hill

(desired number of seedlings = 5 per hill)*
(nominal index length = 9.5 mm; end roller diameter = 25 mm)

stroke No.	connecting pin relative approach position (mm) (measured from the end roller centre line) 0	1	2	3	4	5	6	7	8
1st	4	5	5	5	5	5	5	5	5
2nd	1	2	3	3	3	3	4	4	4
3rd	1	1	0	0	0	0	0	1	1
4th	4	4	4	3	3	3	3	2	2
5th	5	5	5	5	5	5	5	5	4

*as connecting pin tranverses end roller (from Appendix C)

3.2 CONSIDERATIONS FOR REDESIGN

The selection of a satisfactory mechanism to accomplish the required motion is sometimes very complicated by the fact that there are many mechanisms which can produce the same required motion. From among the different selections, two will be presented as a solution to the problem. The first solution involved the application of four bar mechanism principles and cam mechanisms with a roller follower. The second solution used a flat-faced follower and associated mechanism to change the direction of the seedling tray.

The redesign, as envisioned in solution one, accomplished seedling tray travel reversals by using a cam mechanism activated by input from the handle assembly through a four bar linkage. In the original design the picker arm oscillated as the operator pushed and pulled the handle. The minimum available angular displacement of the picker arm assembly was 52 degrees for the shallowest planting depth. At the deepest plant-

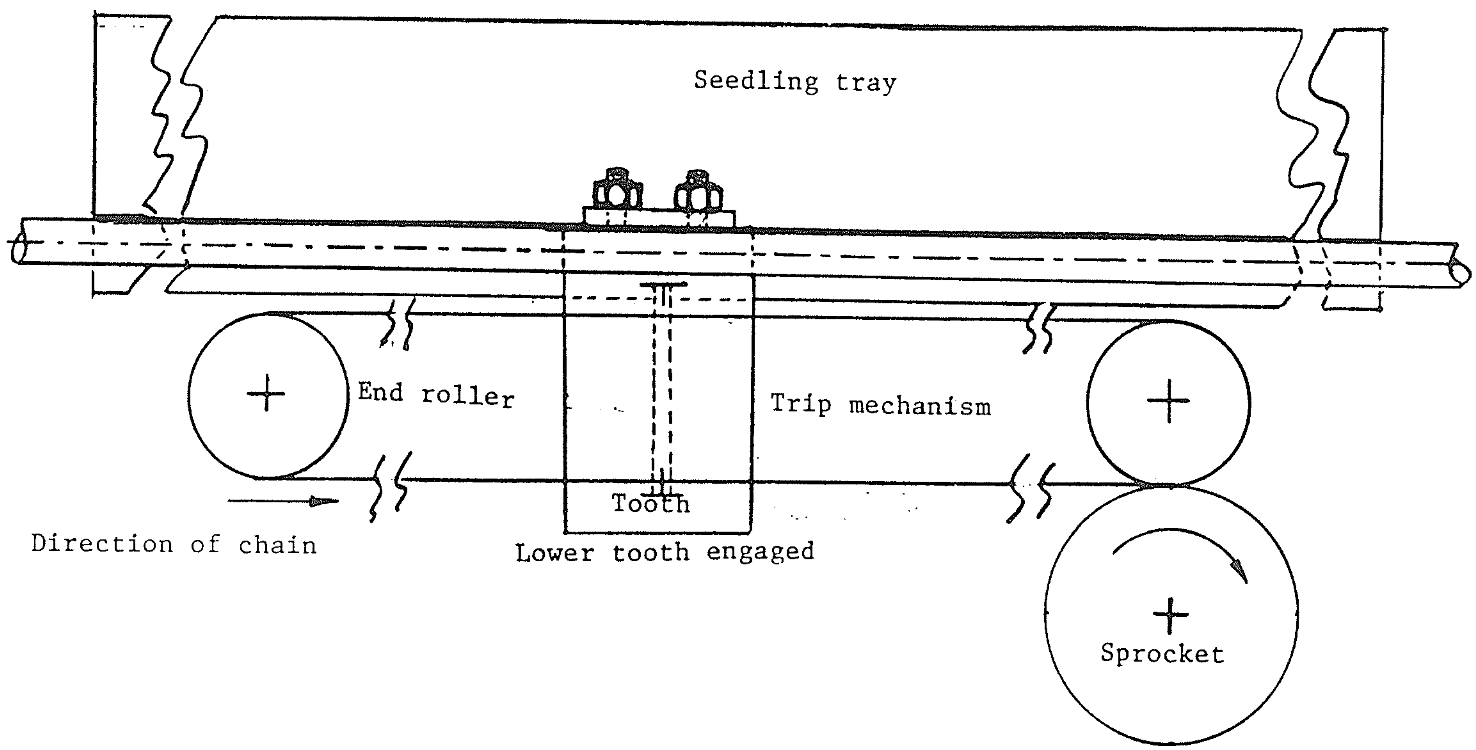


Figure 5: Line drawing of redesigned indexing mechanism

ing depth, the maximum obtainable input angle was 55 degrees. The initial angular position of the oscillating arm relative to the horizontal on the pull stroke was 22 degrees and the final position was 74 degrees. The maximum useable input link length was in the range of 35 to 130 mm. The horizontal distance from the centre of rotation of the oscillating arm to a point directly below the roller chain was 510.0 mm (see Figure 6).

The first design modification was to use a 12 mm diameter shaft on which a tube with a cam was fixed somewhere close to the roller chain. For the maximum output angle, the shortest output link length must be 35 mm.

3.3 DESIGN OF THE FOUR BAR LINKAGE

The four bar linkage was designed using the basic principles of four bar linkage theory. The base requirements of a cam stipulate that for a particular rise the steepness of the cam profile can be reduced by increasing the cam angle (see Figure 6). The maximum available output angle was determined by varying the length of the input link. Since this mechanism was required to transmit motion, the transmission angle was less important. But to avoid than the required cam angle. But to avoid toggle positions the optimum transmission angle was considered in determining the best combination of linkages. An output angle greater than 120 degrees was considered necessary. For this output requirement, the difference in the initial and the final transmission angles must not be greater than 2 degrees.

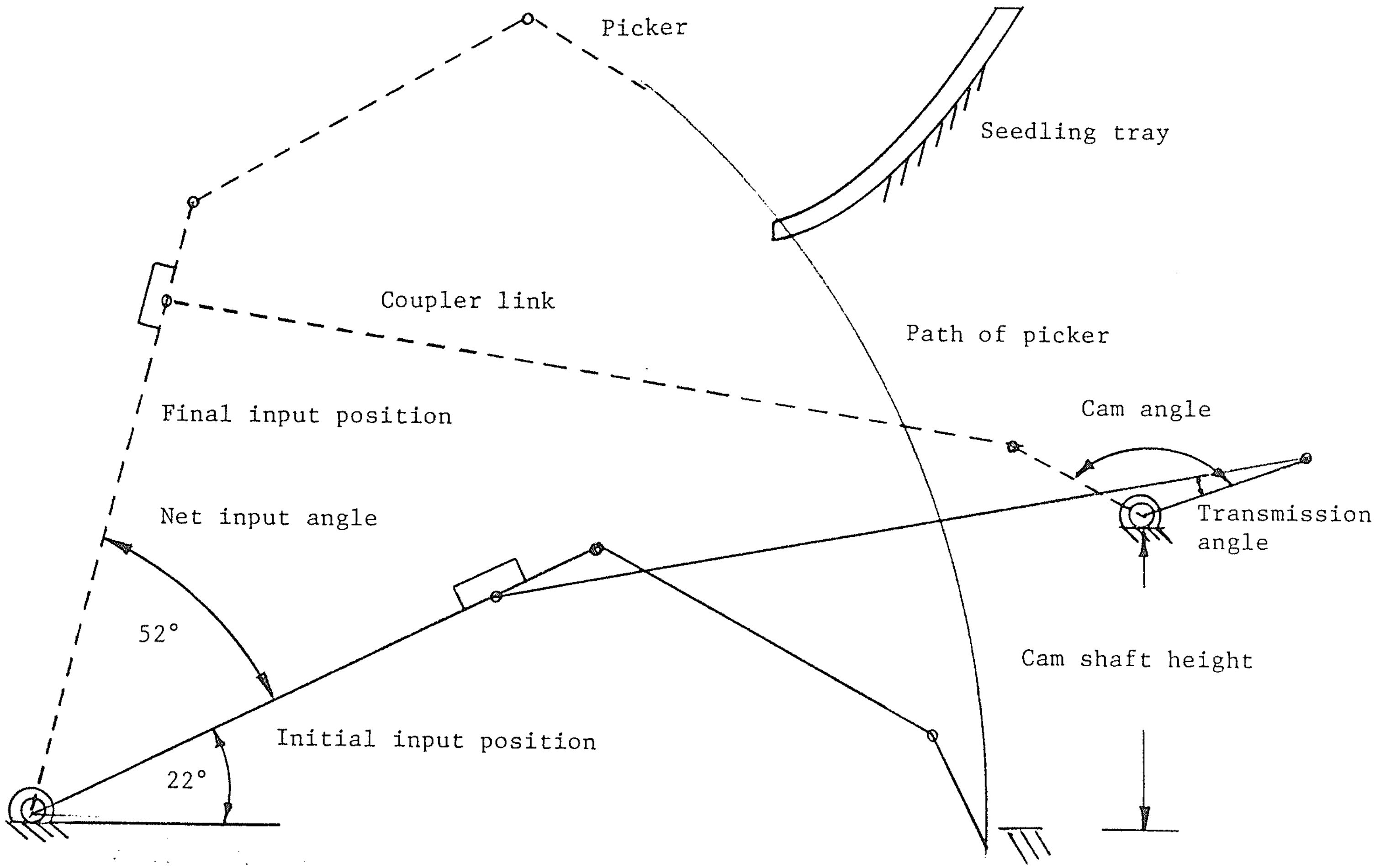


Figure 6: Line drawing to show picker arm and four bar positions

3.4 DESIGN OF THE TRIP MECHANISM

Based on space availability the tentative trip mechanism design lengths were assumed as shown in Figure 7 and 16 (see page 83). The design consisted of follower stem, engaging teeth, a compression spring and a locking device. In designing the trip mechanism, the maximum allowable pressure angle that could be used without jamming the follower stem in its guides was the design criteria. Depending upon the type of follower, the maximum allowable pressure angle will vary. The maximum allowable pressure angle can be calculated as (refer to Figure 7),(Rothbart, 1956 and Chen, 1982)

$$a(\max) = \tan^{-1}[(B)/\{U1(2A+B)\}] - \tan^{-1}(U2) \qquad [3.4\text{-}1]$$

where,

a = pressure angle, degrees

U1 = coefficient of friction between follower stem and guide

U2 = coefficient of friction between follower and cam

B = guide length, mm

A = maximum overhang length, mm

In the equation a, is a function of A and B. For a low speed cam with roller followers, sliding is not considered and pure rolling may be assumed. The value of U2 was assumed to be zero. For flat-faced follower sliding action becomes dominant and the maximum allowable pressure angle will be decreased. However, the pressure angle transmitted through the cam is almost zero regardless of the cam size. Since friction depends

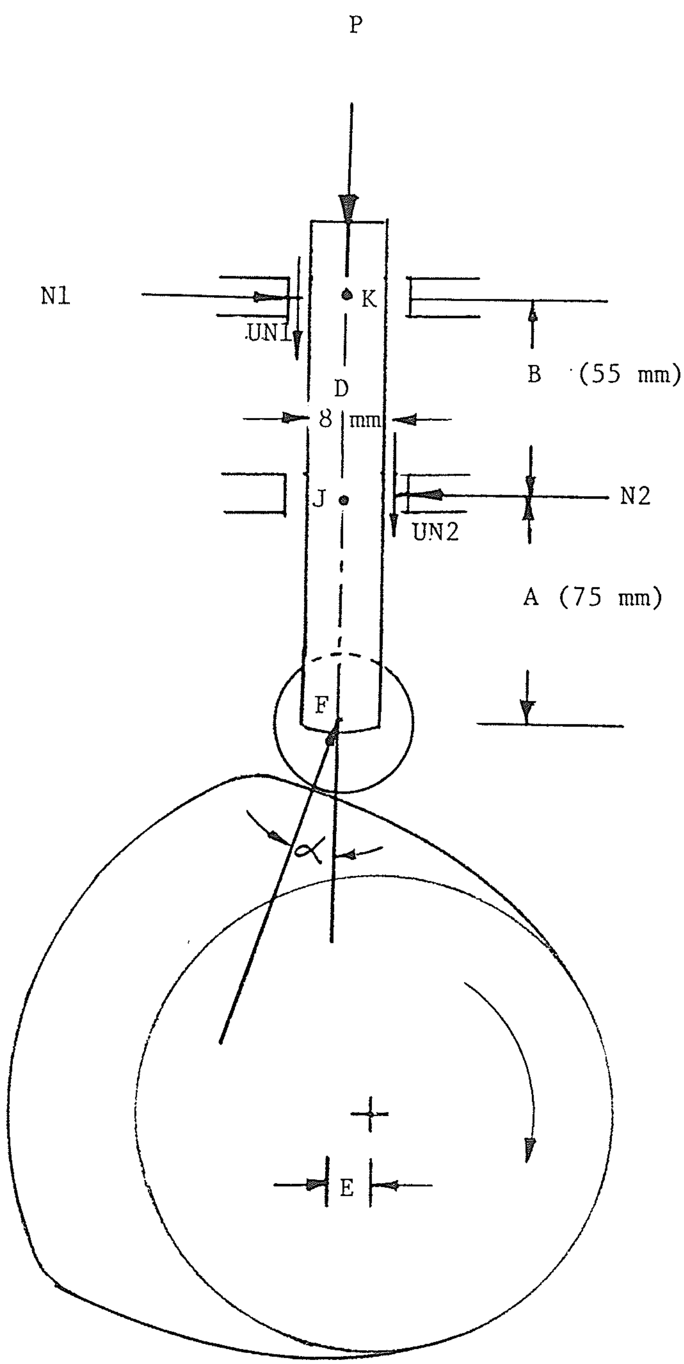


Figure 7: Determination of maximum allowable pressure angle
the guides (not to scale)

upon the roughness of the surface in contact the value of U2 may be high without proper lubrication. The net result will be a reduction in the maximum allowable pressure angle. In the case of a roller follower the value of U2 may be neglected for slow speed cams. But U1 in the guides could be as high as 0.25 if there was backlash in the system.

3.5 SELECTION OF THE FOLLOWER TYPE

The followers in general may be classified in one of three ways:

1. The construction of the surfaces in contact, e.g. knife-edge, roller or flat-faced.
2. The type of movement, whether translating or oscillating.
3. The location of the line of movement with reference to the cam centre, such as radial or offset.

The type of surface contact can be further described as follows:

1. Knife-edge follower: Its construction is simple and the manufacturing cost is cheap. However, due to rapid wear of the surfaces, the use of knife-edge follower is usually avoided.
2. Flat-faced follower: The pressure angle for this type is always zero and the only tendency for jamming of the follower in its guide is due to the moment of the normal force acting on the moment arm and to the frictional force acting on the follower.
3. Roller follower: The roller follower is primarily used where high relative velocities exist between the cam surface and the contacting bodies. In addition, roller followers are used where large loads are to be transmitted. The use of a roller follower

minimizes wear and friction at the point of contact. The disadvantage of the roller follower is that a steep cam tends to jam the translating follower. A larger cam size will eliminate jamming. Another problem can be undercutting. This problem can be eliminated by reducing the size of the roller. In this research these conditions were avoided and the smallest cam size was chosen by offsetting the follower.

In this application, both flat-faced and roller followers are feasible. However, they each have their merits that determine their application. The final choice of the follower type will be an intelligent compromise by the designer.

In using a roller follower, the size of the roller, its weight, the percentage of sliding action, the pressure angle and the roller pin must be considered. In every cam design with a roller follower, it is impossible to have pure rolling. Although most cams are operated under constant angular velocity their peripheral speeds are not always constant. The amount of sliding will vary with changes in the peripheral velocity.

The amount of sliding will be interrelated to the magnitude of the maximum shear stress developed near the cam surface. According to experimental results, pure sliding will have less maximum shear stress developed, whereas pure rolling will have intermediate amounts, and rolling combined with sliding will have the greatest amount (Chen, 1982). The magnitude of the maximum shear stress developed will determine the life of the cam. Therefore, the least sliding action is desirable.

Sliding occurs when the frictional torque at the contact point is less than the torque developed at that point. To reduce sliding the torque developed should be less than the frictional torque. Mathematically,

$$T = Ia$$

$$Ia < U(F)(Rf) \qquad [3.5\text{-}1]$$

where,

T = torque developed between cam and roller N-m

I = moment of inertia of roller assembly, $N\text{-}m\text{-}s^2$

a = angular acceleration of roller, rad/s^2

U = coefficient of friction between cam and roller

F = normal load, N

Rf = radius of the roller, m

As indicated by the general equation [3.5-1], an increase in U, F and Rf will decrease sliding. However, an increase in U and F will increase stress and consequently wear and is therefore undesirable. An increase in Rf (roller follower radius) will result in undercutting problems requiring more space which may be impractical. Reduction in the moment of inertia and the angular acceleration could be possible. Moment of inertia is related to the mass of the roller and therefore the roller mass should be kept low. To reduce the angular acceleration of the roller, the peripheral speed changes of the cam should be kept low.

3.6 SELECTION OF THE FOLLOWER MOTION

Selection of the follower motion was influenced by the speed of the cam and the mass and motion of the follower. Theoretically, rotary cam speeds are not uniform. But their nonuniformity is not appreciable. Therefore, it is the usual practice to assume that the speed of a rotary cam is constant. However, the peripheral speed of the cam is not necessarily uniform except for dwell positions.

The lift or fall of the cam is the net difference between the minimum and maximum radius of the cam profile. The lift and the cam angle are fixed for a particular mechanism. If the speed of the cam is too high, the time required to lift or lower the follower will be short resulting in high speed for the follower.

Changes in velocity together with mass will result in force. Large changes in velocity produces large forces. If the cam speed is high, the time taken to complete a specific angle will be short resulting in large forces. The actual forces developed will not be considered in this research.

Since the force is the product of mass times the rate of change of velocity and since the mass of the follower was kept low, the magnitude of the resulting forces was not too large. Another reason for ignoring the forces was that the follower return was not cam activated. The follower return was actuated by releasing potential energy stored in a spring during the cam lift. If the follower was to be returned by cam action, then the forces should be taken into account. If the spring force and the gravity force are not enough to overcome the inertia forces the follower will not describe the true path of motion as designed. Jerk should not be ignored.

At very low speeds, almost any type of follower motion is suitable. For medium speed cams three types of motion (constant acceleration, simple harmonic or cycloidal) are frequently used. At high speeds, cycloidal, modified sinusoidal, and other types of motion are employed. Rothbart (1956) suggests that with a perfectly rigid member having no backlash or clearance in the system, the constant acceleration curve should be used for best performance. It was necessary to classify the speed of the cam used in this study and the assumption of constant angular velocity was practical.

In this research the cam was considered as a rotary type because every point on the cam rotated about the same fixed centre. But the cam shaft did not complete one revolution since it oscillated between 0 and 144 degrees. The motion event was "Dwell, Rise, Dwell". The number of degrees for each motion event were calculated based on the requirements of the mechanism. All possible input angles of the picker arm could not be accounted for. Some allowance had to be made for adjustment of planting depth. The initial dwell angle allowed for the adjustment of depth. The final dwell was to compensate for lack of rigidity in the linkage. Therefore, the first 27 degrees were used for dwell followed by 100 degrees for rise and a final 17 degrees for dwell making the total cam angle 144 degrees.

The total lift of the follower was 14 mm. The follower was lifted this amount for a cam rotation of 100 degrees. The cam was initially at rest. The times required to turn the cam to the 27 and 127 degree positions were observed and recorded as 1/10 and 1/2 second, respectively. The angular velocities at 27 and 127 degree were estimated to be 5 rad/s. The estimate of the equal velocities was based on observing an operator work the handle. Although the initial velocity might have been faster, the overall velocity was rather uniform. The intermediate angular velocity between the 27 and 127 degree positions could be slightly different. In designing the cam the assumption of uniform angular velocity was made. This was reasonable and very nearly true.

In classifying the speed of a cam as high, medium or low, there are no standard ranges. Cam speed is usually considered medium if the mechanism involves inertia forces but is considered high speed if both inertia and vibration must be considered. For other less severe conditions, the cam speed is considered low. In this design, the follower had negiligible mass with only small spring pressure. For consideration of the durability of welded joints and adjustable joints in the trip mechanism the cam was treated as medium speed.

The basic motion of followers is primarily modeled by simple polynomial or trigonometric functions. Each motion has its own characteristics. To visualize the nature of the individual motion, it is better to describe displacement, velocity, acceleration and jerk equations as a function of cam angle. Plotting these values for the abcissa and the corresponding cam angles for the ordinate illustrates the cam action. The displacement equation when modeled by a simple polynomial may be represented as ($Y' = dy/d\emptyset$, etc.):

$Y = f(\emptyset)$ (displacement of follower, mm)

$$Y' = f'(\emptyset) \qquad \text{(velocity, mm/degree)}$$

$$Y'' = f''(\emptyset) \qquad \text{(acceleration, mm/degree}^2\text{)}$$

$$Y''' = f'''(\emptyset) \qquad \text{(jerk, mm/degree}^3\text{)}$$

In general:

$$Y = C(\emptyset)^n \qquad [3.6\text{-}1]$$

where,

Y = displacement of follower, mm

C = constant

$\emptyset$ = cam angular movement, radian

n = any number

when, $n = 1$ the general equation becomes

$$Y = C(\emptyset)$$

where,

$Y(max) = H$, the maximum rise, mm and $\emptyset = \beta$ is the maximum cam angle, rad.

It follows that $C = H/\beta$, so that

$$Y = (H/\beta)\emptyset \qquad [3.6\text{-}2]$$

Equation [3.6-2] represents straight line motion or constant velocity motion. By successive differentiations the velocity, $Y' = H/\beta$, the acceleration, $Y'' = 0$, and the jerk, $Y''' = 0$, are obtained. These relationships are illustrated in Figure 8. From the diagram, it can be seen that the acceleration is zero except at the begining and the end of the rise where the acceleration is theoretically instantaneously infinite. This acceleration transmits a high shock to the follower linkage the magnitude of which depend on the linkage flexibility. For a Dwell-Rise-Dwell cam this motion is therefore not practical.

For constant acceleration, n in the general equation [3.6-1] is 2. By an analysis similar to that above, expressions for displacement, velocity, acceleration and jerk can be derived. The resulting displacement, velocity, acceleration and jerk are illustrated in Figure 9. This motion is termed parabolic. The diagram illustrates positive and negative accelerations. But the acceleration's maximum absolute magnitude is the lowest for a given motion. However, there is an abrupt change in acceleration at the begining and end and at the transition point. This is undesirable for high or even medium speed applications.

Trigonometric curves are based on simple harmonic motion, cycloidal motion and elliptical motion. For medium speed cams consideration of simple harmonic motion and cycloidal is acceptable (Rothbart, 1956).

Simple harmonic motion and cycloidal motion can be analyzed as above. The displacement, velocity, acceleration, and jerk diagrams for simple harmonic motion and cycloidal motion are shown in Figures 10 and 11, respectively.

Simple harmonic motion has smooth continuous acceleration with sudden changes only at the begining and end of dwell. As a consequence the jerk becomes infinite at these points. However, this type of motion has low follower side thrust, smooth starting and reasonable follower spring size compared to the basic polynomial curves.

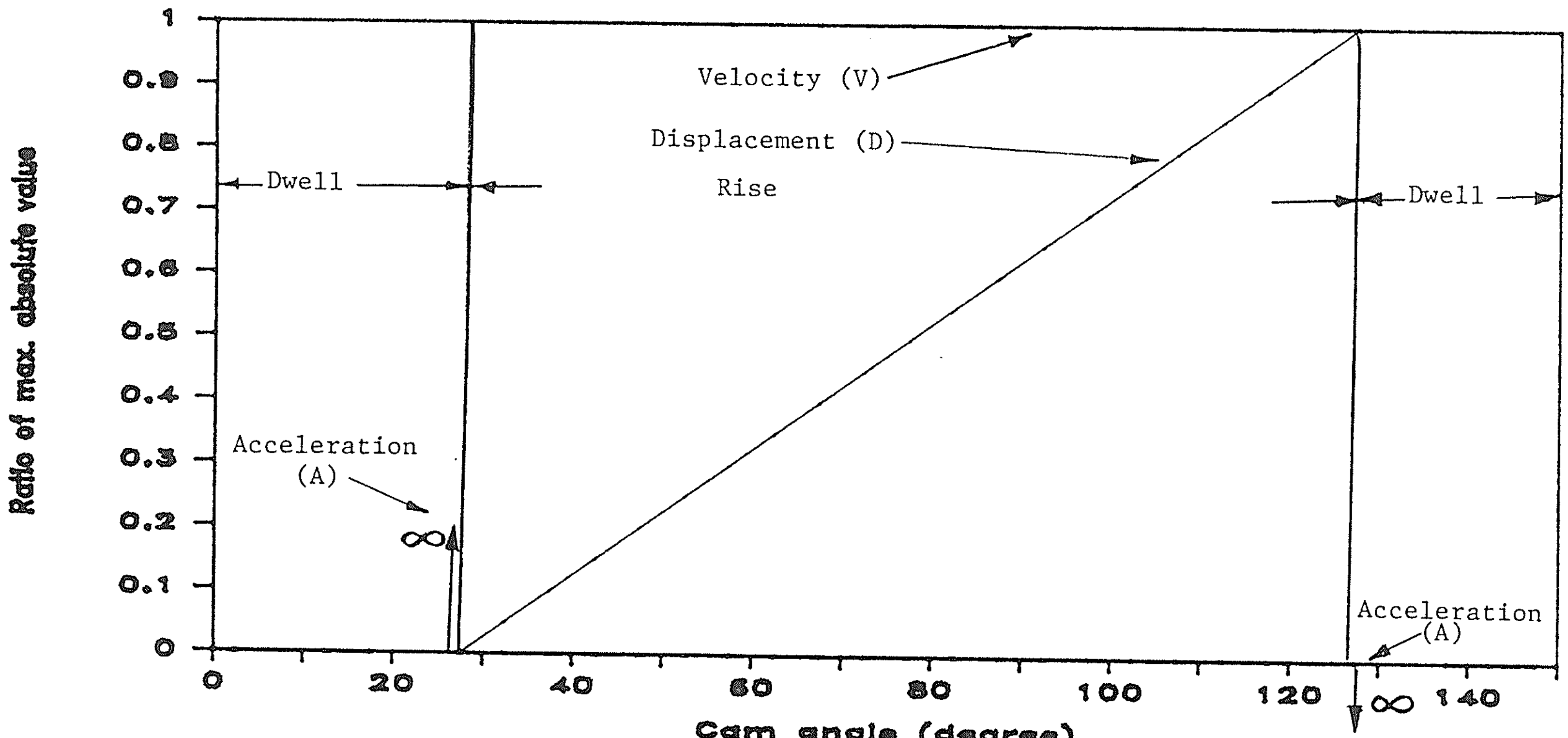


Figure 8: Motion diagrams for constant velocity motion of cam action

(DMX=0.014 m; VMX=0.12 m)

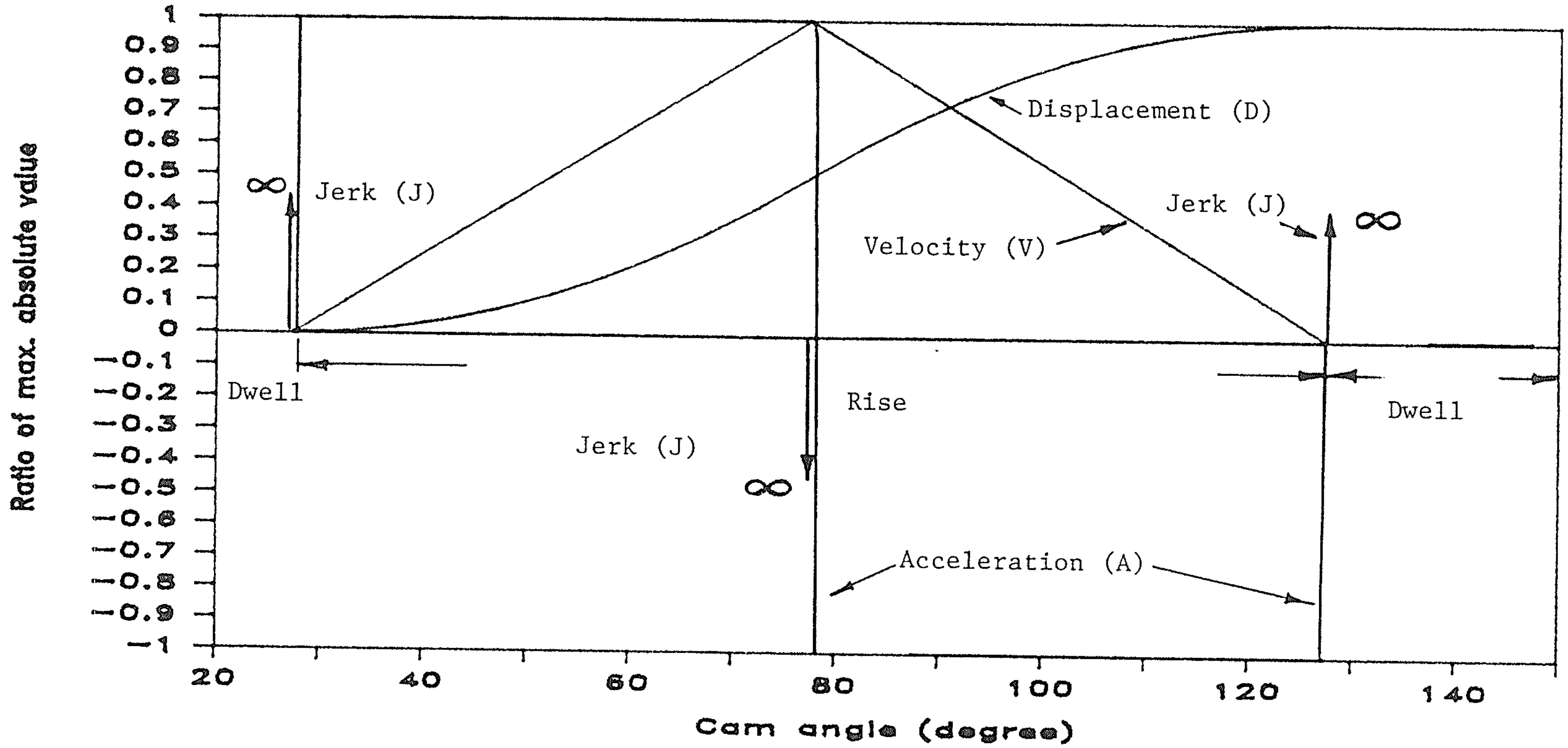


Figure 9: Motion diagrams for parabolic motion of cam action
(DMX=0.014 m; VMX=0.08 m/s; AMX=0.45 m/s^2)

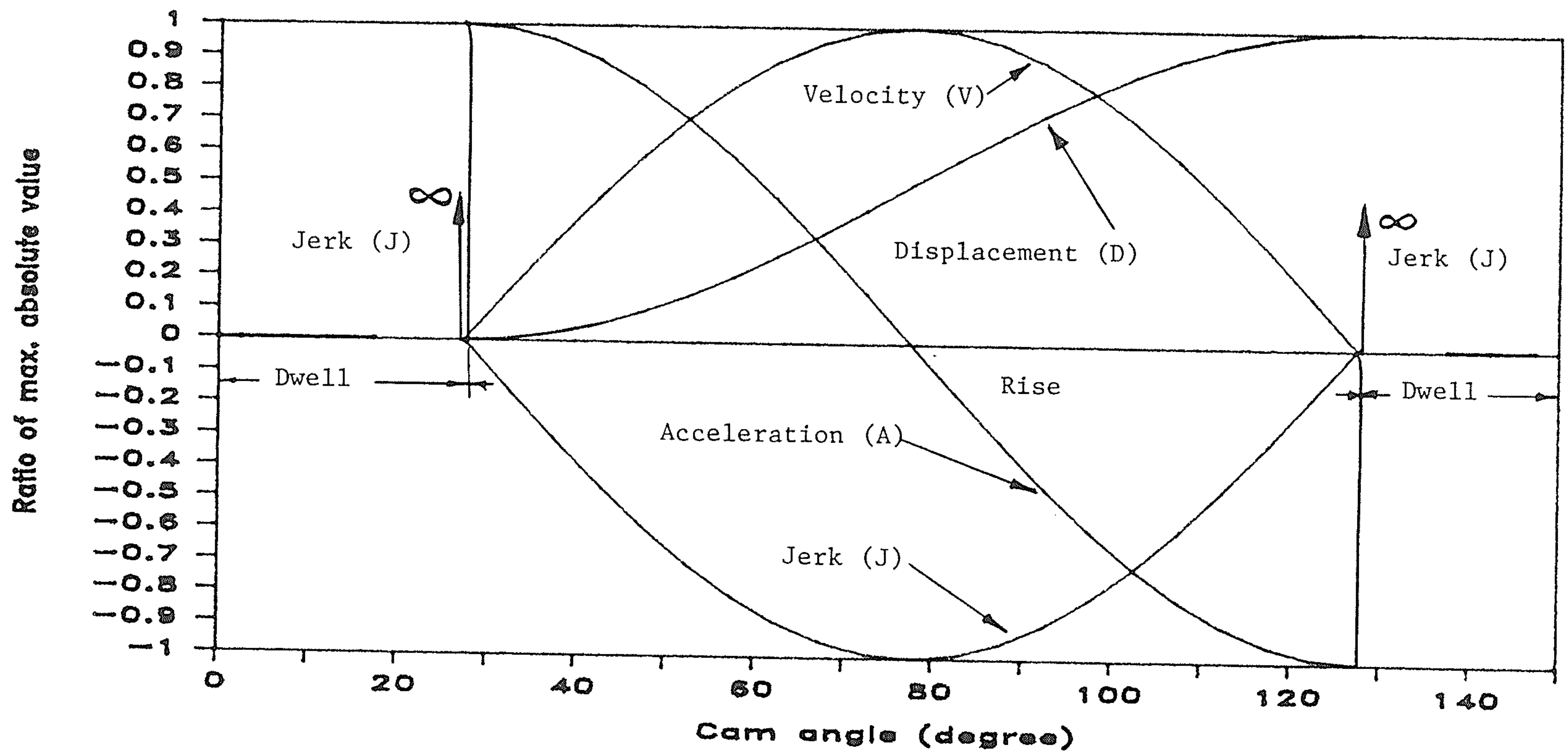


Figure 10: Motion diagrams for simple harmonic motion of cam action (DMX=0.014 m; VMX=0.06 m/s; AMX=0.56 m/s^2; JMX=5.1 m/s^3)

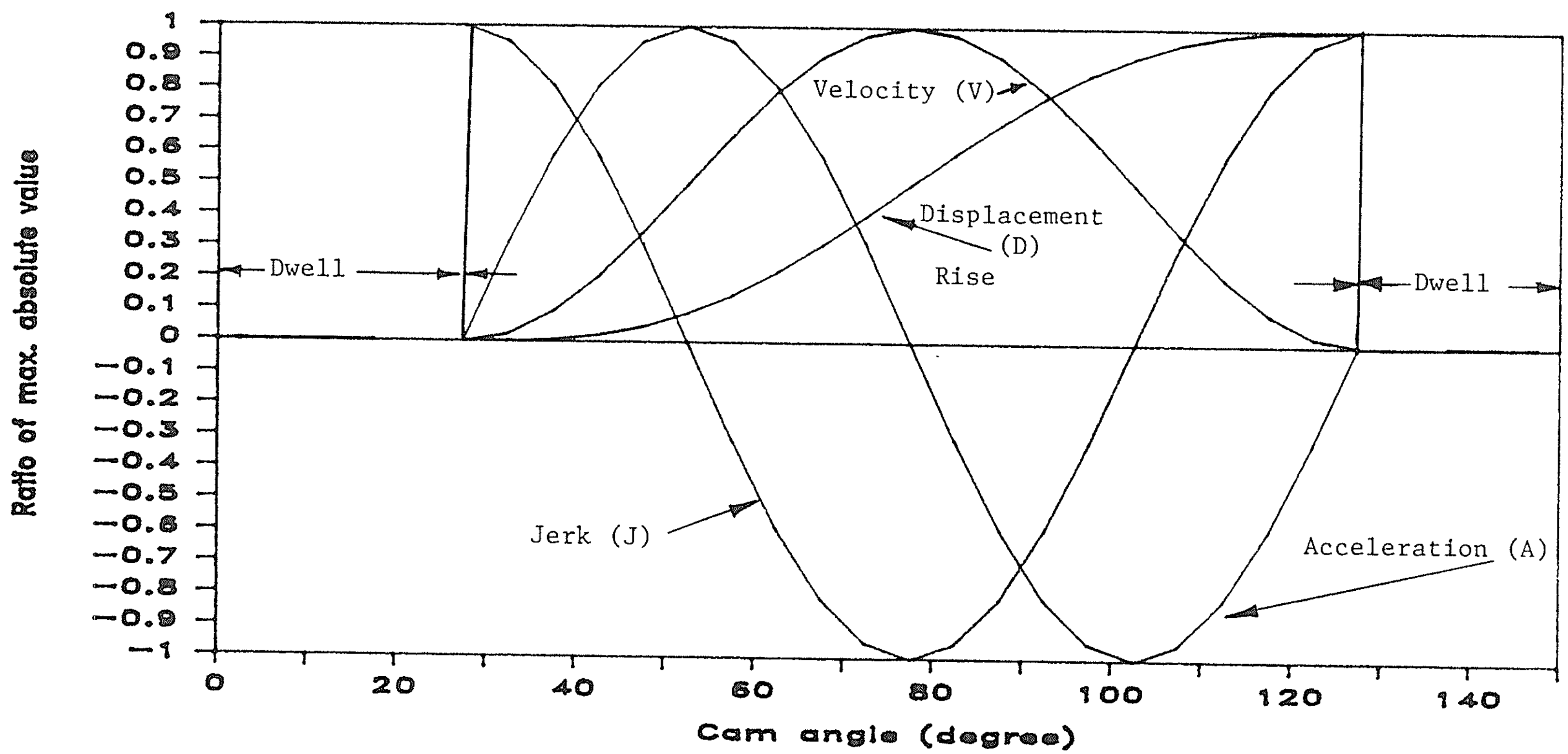


Figure 11: Motion diagrams for cycloidal motion of cam action
(DMX=0.014 m; VMX=0.08 m/s; AMX=0.72 m/s^2; JMX=13 m/s^3)

For cycloidal motion, there is no sudden change in acceleration at the end of the dwell. The cycloidal cam has the smoothest motion of all of the basic types of motion. But the magnitude of the acceleration is greater than that of the constant acceleration motion.

In selecting a suitable follower motion, the acceleration characteristic should be carefully examined since the acceleration determines the inertia force. Inertia force is the product of mass and acceleration. Large acclerations will not lead to large inertia forces provided that the mass of the follower is negiligible. Therefore, when selecting a follower motion, the mass of the follower must be considered. A sudden change in acceleration means a large value of jerk resulting in a sudden change of the force on the follower. This shock effect is very undesirable for the mechanism.

According to theory, infinite acceleration should cause infinite force. Thus, although elasticity and friction in the mechanism have cushioning effects, the extremely high force will damage the mechanism unless the cam turns very slowly. Therefore, although parabolic motion has the smallest relative acceleration for a specified displacement during a given time interval, infinite jerk at the transition point precludes its use.

With perfectly rigid linkage members, having no backlash or clearance in the system, the constant acceleration motion will give excellent performance (Rothbart, 1956). Moreover, inertia forces which are proportional to the acceleration are easily determined. However, all linkage members are somewhat elastic and clearance and backlash always exist so parabolic motion is not desirable.

Cycloidal motion for most machine requirements is the best selection. Since cycloidal motion has no abrupt change in acceleration the motion gives the lowest vibration, wear, stress, noise and shock. Motion initiation is easy. Only small springs are needed to prevent backlash. For these reasons cycloidal motion was selected. Maximum absolute velocity, acceleration and jerk for the respective motion are shown in Table 4. Table 5 shows ratios for the relative magnitude.

TABLE 4

Maximum absolute velocity, acceleration and jerk

for the respective motion of follower rise = 0.014 m; cam angle = 100 degrees; angular velocity = 5 rad/s

Type of motion	Velocity m/s	Acceleration m/s^2	Jerk m/s^3
Constant Velocity	0.04	-	-
Parabolic Motion	0.08	0.46	-
Simple harmonic	0.06	0.56	5.1
Cycloidal Motion	0.08	0.72	13.0

TABLE 5

Comparison of maximum absolute magnitude ratios

(relative to the maximum for each motion type)

Type of motion	Magnitude of		
	Velocity	Acceleration	Jerk
Constant Velocity	1	-	-
Parabolic Motion	2	1	-
Simple harmonic Motion	1.58	1.23	1
Cycloidal Motion	2	1.57	2.54

3.7 DESIGN OF CAM SIZE

Having decided that the follower motion should be cycloidal it was then necessary to determine the cam dimensions. A key concept in cam design is to make the cam as small as possible. Larger cam sizes are undesirable. Large cams require more precise cutting points in manufacturing and are, therefore, more expensive to make. Larger cams also have high circumferential speeds and any small deviations from the theoritical path of the follower causes additional acceleration which increases as the square of the cam size. The inertia of large cams may interfere with quick starting and stopping.

If the pressure angle is not critical, a small cam size is desirable since for a given required motion the smallest cam size possible will result in the least side thrust reversal and consequently quieter operation. Maximum cam size is limited by the allowable pressure angle except in the case of a follower with a flat-face. For the flat-faced follower, the pressure angle is always constant and is almost zero and is therefore not limiting in most designs. Jamming of the follower in the guide with a flat-faced follower is not because of the pressure angle but because of the moment created by the force at the contact point of the follower with the cam. Jamming related to pressure angle is similar. Consequently, the width of the flat-faced must be at least as great as double the maximum moment arm so that the follower moves exactly as required.

The maximum moment arm can be determined from a cam drawing. The cam motion at which maximum moment arm occurs can be easily determined from a displacement diagram. On the displacement diagram the angle corresponding to the maximum absolute slope locates the maximum moment arm. On a graphical lay out the length can be measured from the angle position perpendicular to the contact point of the cam on the left side if the rotation of the cam is clockwise or to the right side if rotation of the cam is opposite.

The maximum moment arm can be determined more easily and accurately by analytic methods. The method is derived from the velocity of the follower. Mathematically, the follower speed may be expressed as:

$$dy/dt = L\, d\phi/dt \qquad [3.7\text{-}1]$$

where,

L = the instantaneous distance of the contact point from the follower stem axis, mm

dy = the small displacement of the follower, mm

$d\emptyset$ = the small angular movement of cam, radians

dt = the corresponding time taken, s

Then,

$$L = dy/d\emptyset$$

and the maximum value of L becomes,

$$Lm = |dy/d\emptyset|max \qquad [3.7\text{-}2]$$

In the computer program the value Lm was represented by YDMAX and was 16.01 mm. Therefore, the minimum width of the flat-faced follower should be 2(Lm). Allowing for a margin of safety, it is suggested that the width of the flat-faced follower be made 2(Lm)+10 mm if space permits. The actual value is dependent on the type of the follower, total cam angle and the rise of the follower. For cycloidal motion, maximum moment arm occurs at the mid stroke of the follower.

With a flat-faced follower the only tendency of the stem to jam in the guides is due to the normal force acting on the moment arm L and to the frictional force on the face of the follower. Lm is related to the overhang length of the follower, the guide length and the friction in the system. For the follower to act without jamming, the length of Lm should be:

$$Lm + E < B/2U1 + U2(1+N)/2 \qquad [3.7\text{-}3]$$

where,

Lm = the maximum length of the moment arm, mm

E = the offset distance, mm

B = the guide length, mm

N = the ratio of overhang length to guide length, unitless

U1 = coefficient of friction between follower stem and the guide

U2 = coefficient of friction between the cam and the flat-faced follower

To satisfy the inequality, jamming can be avoided by reducing the left hand term Lm or by increasing the right hand term. In the left hand term Lm is a function of the type of follower motion, the maximum rise of the follower and the total cam angle. The rise and the type of motion are a basic requirement of the mechanism. Therefore, they cannot be varied or replaced. The only way to reduce the value of Lm is to increase the cam angle. But the cam angle has a restricted range of values in this design. Generally, the cam angle in other designs would have constraints also. An increase in the cam angle would not appreciably reduce Lm.

For the right hand term, either an increase in the guide length or in the ratio N would increase the whole term. However, an increase in the guide length alone will increase the whole term appreciably. On the other hand, an increase in the overhang length, keeping the guide length constant, will increase the right hand term. Since this action acts through the coefficient of frictional resistance, the result is not very noticeable. In addition, other undesirable side effects can be expected. Once the flat face wears, then the normal to the cam surface is not parallel to the line of motion with the result that the pressure angle is no longer zero and then the maximum allowable pressure angle of the trip mechanism could become critical. Moreover, the maximum allowable pressure angle will be further reduced by an increase in the overhang length. An increase in the guide length is desirable because if a flat-faced follower was to be replaced with a roller follower the design conditions will still be satisfactory.

Friction in the system can be reduced using lubricants, bearings or appropriate materials with smooth contacting surfaces. Rothbart (1956) suggested that the friction in the guides should be assumed to be 1.5 times the actual value, or 0.25. Heavy loads associated with low speeds and lack of rigidity might increase actual friction by a factor of 1.5 over experimentally determined static friction values (Chen, 1982). In this application, the use of lubrication and ball bearings to reduce the friction is not possible. At the same time, backlash in the system is most likely to be present. Therefore, taking the value of U1 to be 1.5 times its normal value will cover all eventualities imposed by the system. If simplification of equation (3.7-3), based on the above postulates, still satisfies Lm then the tentative design values of B and N can be used safely.

3.8 DEVELOPMENT OF THE CAM PROFILE WITH FLAT-FACED FOLLOWER

In the case of a cam with a flat-faced follower the size of the cam is not only limited by pressure angle. It could also be limited by cusps or undercutting. The problem occurs when all tangents drawn at each position of the cam profile do not touch the flat-faced follower. The cam thus developed is incapable of driving the follower in the desired position. The problem cam be avoided by increasing the radius of the cam.

Larger cam sizes are undesirable so the minimum possible radius of the cam that is free from cusps needs to be determined. Using a graphical method the points of contact between the follower and the cam are unknown so the minimum radius of the cam to be free from cusps is determined by trial. The exact solution can also be obtained by an analytical approach. Referring to Figure 12, the general equation of a cam with a flat-faced follower can be written as:

$$Y = mX + C \tag{3.8-1}$$

where, slope $m = -\cot \emptyset$, and intercept $C = R/\sin \emptyset$. But, $R = Rb + f(\emptyset)$ where, Rb = the base circle radius of of the cam, mm, and $f(\emptyset)$ = the lift of the cam, mm from the lowest position to the point where the angular movement is considered. It follows that,

$$Rb + f(\emptyset) = X \cos \emptyset + Y \sin \emptyset \tag{3.8-2}$$

Equation (3.8-2) represents a family of straight lines giving all positions of the flat-faced follower. Equation (3.8-2) is a function of X,Y,Ø, and functions of X,Y and Ø From equation (3.8-2), by rearranging, the following is obtained:

$$X \cos \emptyset + Y \sin \emptyset - Rb - f(\emptyset) = 0 \tag{3.8-3}$$

To solve for X and Y, differentiate equation (3.8-3) with respect to Ø and rearrange to give:

$$f'(\emptyset) = Y \cos \emptyset - X \sin \emptyset \tag{3.8-4}$$

Then solving equations (3.8-2) and (3.8-3), gives:

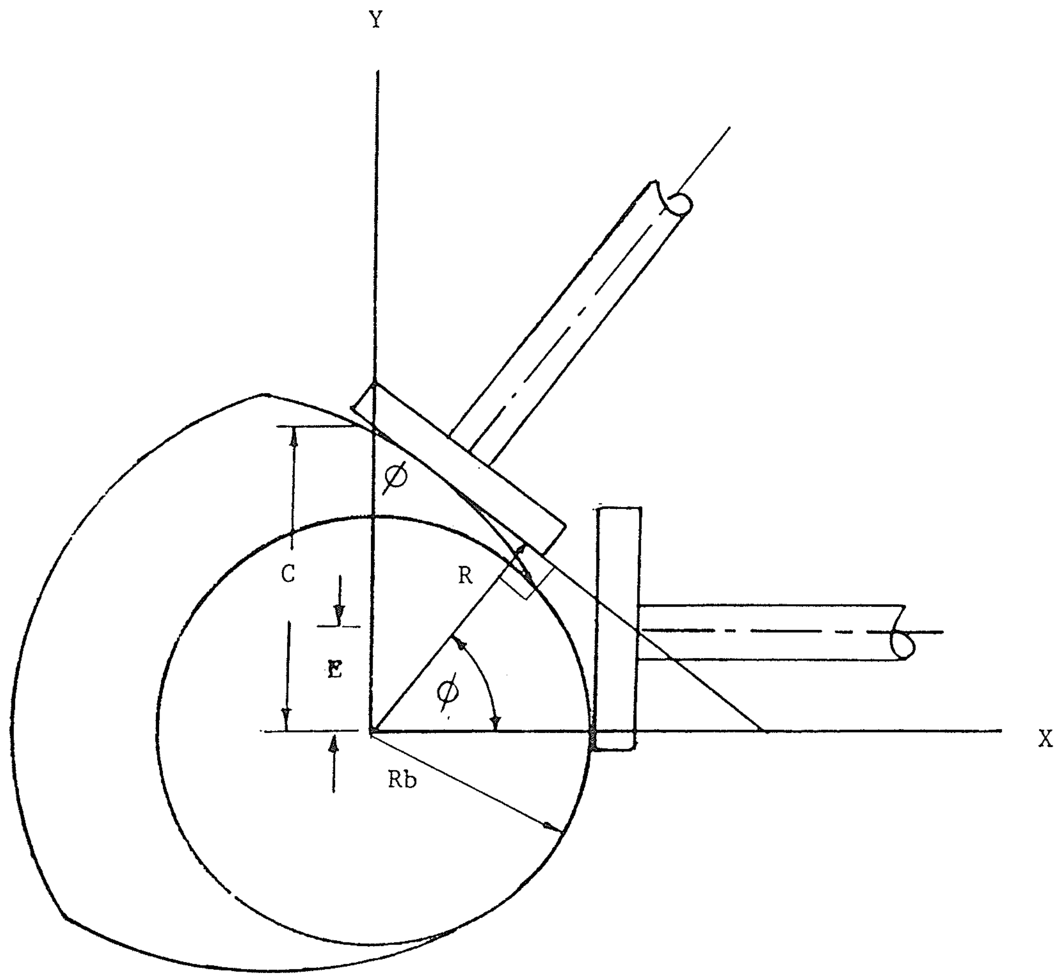


Figure 12: Determination of cam size with flat-faced follower

$$X = [Rb+f(\emptyset)]\cos \emptyset - f'(\emptyset)\sin \emptyset$$

[3.8-5]

$$Y = [Rb+f(\emptyset)]\sin \emptyset + f'(\emptyset)\cos \emptyset$$

[3.8-6]

X and Y are the coordinates of the required cam profile.

Rb is the base radius of the cam and has a certain minimum value for the cam to be free of cusps. Cusps occurs when $dX/d\emptyset = 0$ and $dY/d\emptyset = 0$, simultaneously.

That is,

$$dX/d\emptyset = -[(Rb+f(\emptyset)+f''(\emptyset)]\sin \emptyset$$
$$= 0$$

[3.8-7]

$$dY/d\emptyset = [Rb+f(\emptyset)+f''(\emptyset)]\cos \emptyset$$
$$= 0$$

[3.8-8]

Since $f(\emptyset)$ and $f''(\emptyset)$ are finite, it is possible to select a value of Rb such that, $Rb+f(\emptyset)+f''(\emptyset) > 0$. This requires that $f(\emptyset)+f''(\emptyset) > -Rb$ for all values of $\emptyset$. The value of Rb satisfying this condition substituted into equations (3.8-5) and (3.8-6) will produce a cam profile for a flat-faced follower that will be free of cusps.

3.9 CAM SIZE WITH ROLLER FOLLOWER

In the case of a roller follower the pressure angle becomes critical. The minimum possible pressure angle and cam size are the main objectives in the design of the cam mechanism. But pressure angle has a negative correlation with cam size. A compromise needs to be made between pressure angle and cam size.

The pressure angle is the angle between the normal at the cam surface and the line of follower motion. Its range is between ±90 degrees. Pressure angle is dependent on the size of the cam, the size of the roller follower, the cam angle, the type of follower motion and finally the offset distance to some extent. Pressure angle can be reduced by increasing cam size. But cam size is sometimes limited by the space availability on the machine. If space is no problem wear conditions should still be checked. A large cam size increases the peripheral speed for the same angular velocity. This could prevent the follower from having pure rolling action.

The resulting sliding action could result in a wear problem. Offsetting could solve the problem. For the specified maximum pressure angle condition the size of the cam can be reduced by offsetting. The offset distance with related reduction in base radius of cam is shown in Figure 13.

Offsetting also has some limitations. It is impossible to offset the follower distance more than the base circle radius of the cam. It is also interesting to observe that offsetting does not decrease the pressure angle within the base circle radius range. Maximum obtainable offset distance is the base radius of the cam. Offsetting is not directly related to the pressure angle within this range. It varies inversely over a certain length and has an optimal distance. The distance depends on the type of follower motion, the cam angle and the rise of the follower. Normally the offset distance is limited by the value of the converted velocity or the base circle radius whichever is the smallest. The pressure angle can only be reduced in one direction at a time. When the

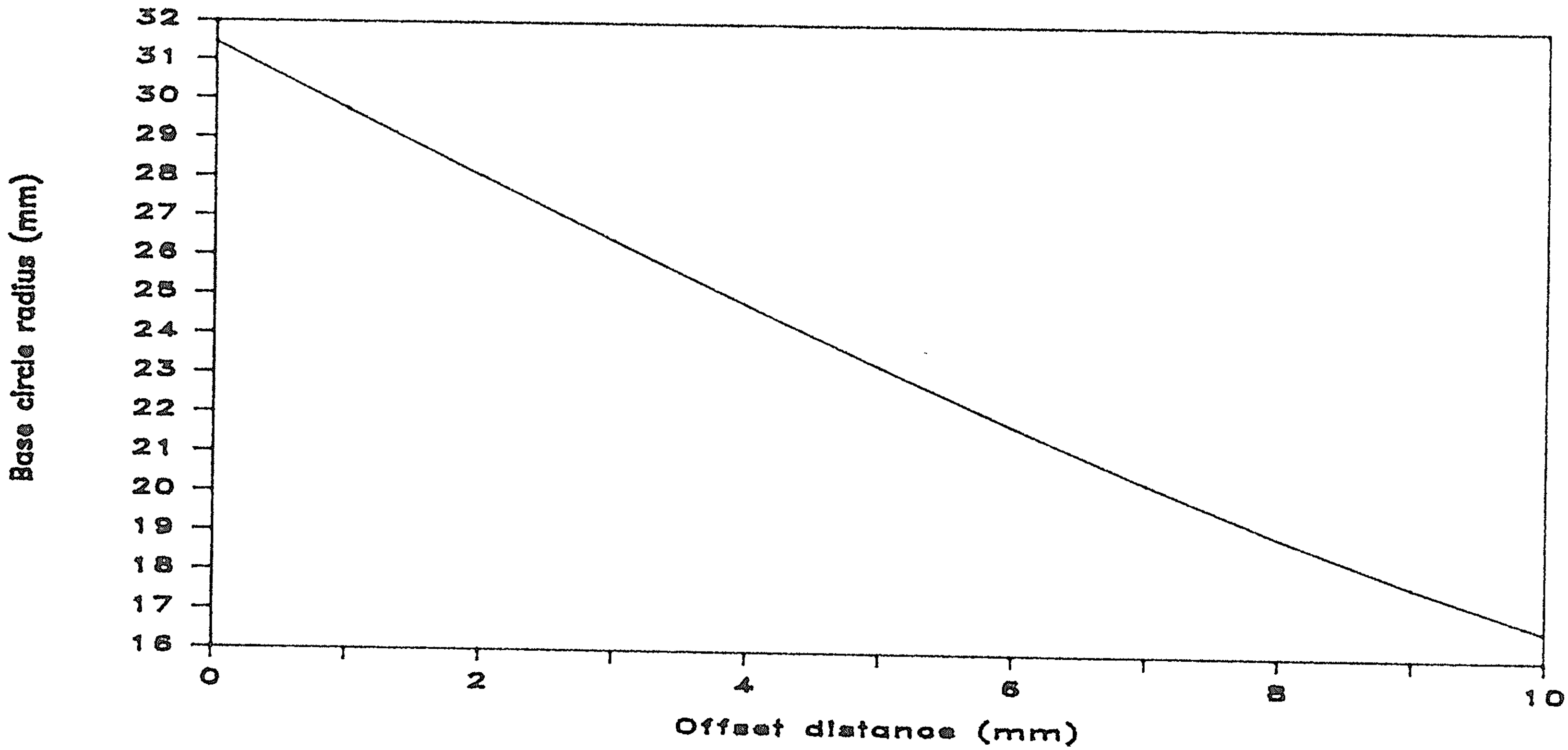


Figure 13: Effect of offsetting on cam size

upward pressure angle is decreased the downward pressure angle will be increased. Cam designers are concerned of jamming which always occurs on the upward motion. To achieve a decreased pressure angle on upward motion proper location of the offset is important. An improper location will increase the pressure angle instead of decreasing the pressure angle. The offset distance should be on the right side if the cam is rotating counter clockwise otherwise it should be on the left side.

The size of the roller follower affects the pressure angle indirectly. The prime circle radius is the sum of roller radius and the base radius of the cam. The prime circle is the minimum working profile of the cam. If the roller radius is almost the same as the prime radius then the base circle radius can be smaller. Therefore, the peripheral length of the cam which has to lift the follower the same amount will be proportionately smaller resulting in steepness at the cam nose. A steep cam nose will develop more stress at the nose causing excessive wear. In addition more power will be needed to lift the follower. These conditions are very undesirable since the pressure angle is also increased. The cam will be harder to operate and there will be more wear in the guides. Even if the size of the cam does not result in excessive values of the pressure angle over the desired range, undercutting should be checked.

The total cam angle and the corresponding rise affect the pressure angle also. To reduce the pressure angle an increase in the cam angle or a decrease in the total follower rise is effective. The net effect is an increase in the length of the path as well as a reduction in the slope. These changes are often not possible because of the basic requirement in the design. Generally, a safe limiting pressure angle in practice is 30 degrees (Rothbart, 1956). The general equation for the pressure angle can be written as:

$$\alpha = \tan^{-1}[(Y' - E)/\{Y + (RO^2 - E^2)^{0.5}\}] \qquad [3.9\text{-}1]$$

where,

Y = follower displacement [or f(Ø)],mm

Y' = derivative of Y with respect to cam angle [or f'(Ø)], mm/degree

E = offset distance, mm

RO = the prime radius of the cam, mm

In the design of the cam size, the pitch point plays an important role. It would be impossible to determine the cam size without determining the pitch point first. To find the pitch point, some parameters must be specified, for example, the follower type and motion and the maximum allowable pressure angle.

Many experts and design books suggest that the maximum pressure angle should be less than 30 degrees (Berard and Waters, 1924). For light loads with accurate, low friction bearings Rothbart (1956) has successfully used pressure angles as high as 47 degrees. In this thesis lack of rigidity in the mechanism and lack of friction bearings or even lubrication limits the pressure angle to no more than 30 degrees. The type of follower motion that has compatible characteristics in this situation is cycloidal. The benefits of offsetting indicate that offsetting can be used to advantage.

Substituting the desired pressure angle in the pressure angle equation [3.9-1] will determine conditions for a particular cam size and offset at that instantaneous position. But the desired maximum pressure angle might be exceeded at some other cam angle positions. The maximum desirable pressure angle must not be exceeded at any position within the cycle of the cam.

To limit the pressure angle to the maximum allowable the method of calculus can be used. The existing equation is differentiated and the result set to zero. The condition obtained is back substituted in the previous general formula. To set the maximum pressure angle condition, differentiation of the pressure angle equation, with respect to time, (equation [3.9-1]) gives:

$$\sec^2\alpha \ d\alpha/dt = d/dt \ (M/N) \qquad [3.9\text{-}2]$$

where,

$$M = Y' - E$$

$$N = Y + (RO^2 - E^2)^{0.5}$$

$$\tan \alpha(max) = Y''(p)/Y'(p) \qquad [3.9\text{-}3]$$

where,

$$Y' = f'(\emptyset)$$

$$Y'' = f''(\emptyset)$$

where p represents the pitch point condition. Applying the condition of cycloidal motion gives:

$$\tan \alpha(\max) = G/H \qquad [3.9\text{-}4]$$

where,

$$G = 2\text{pi}[\sin(2\text{pi}\ \emptyset/\beta)]/\beta$$

$$H = [1-\cos(2\text{pi}\ \emptyset/\beta)]$$

Equation [3.9-4] is the required condition for limiting the pressure angle to the specified maximum. An attempt to determine the required pressure angle in equation [3.9-4] leads to an unmanageable transcendental equation. To determine the pitch point angle explicitly the problem was solved using a high speed computer (see Appendix D).

The pitch point thus obtained determines the size of the cam. This cam size can be further reduced by offsetting over the range of maximum and minimum converted velocity for the specified motion. An offset distance with an increment of 1.0 mm was used to calculate the corresponding cam size. The cam size so determined was not necessarily the final solution. Due to the nature of the tangent function the angle determined to satisfy the pitch curve will have two values. Since both angles can not be used the correct one must be selected. Using an iteration method, the optimum offset distance was determined for the smallest possible cam size. For different cam angles of the cam used in the redesign the pressure angle is illustrated in Figure 14. It can be seen that no maximum absolute pressure angle is greater than 30 degrees.

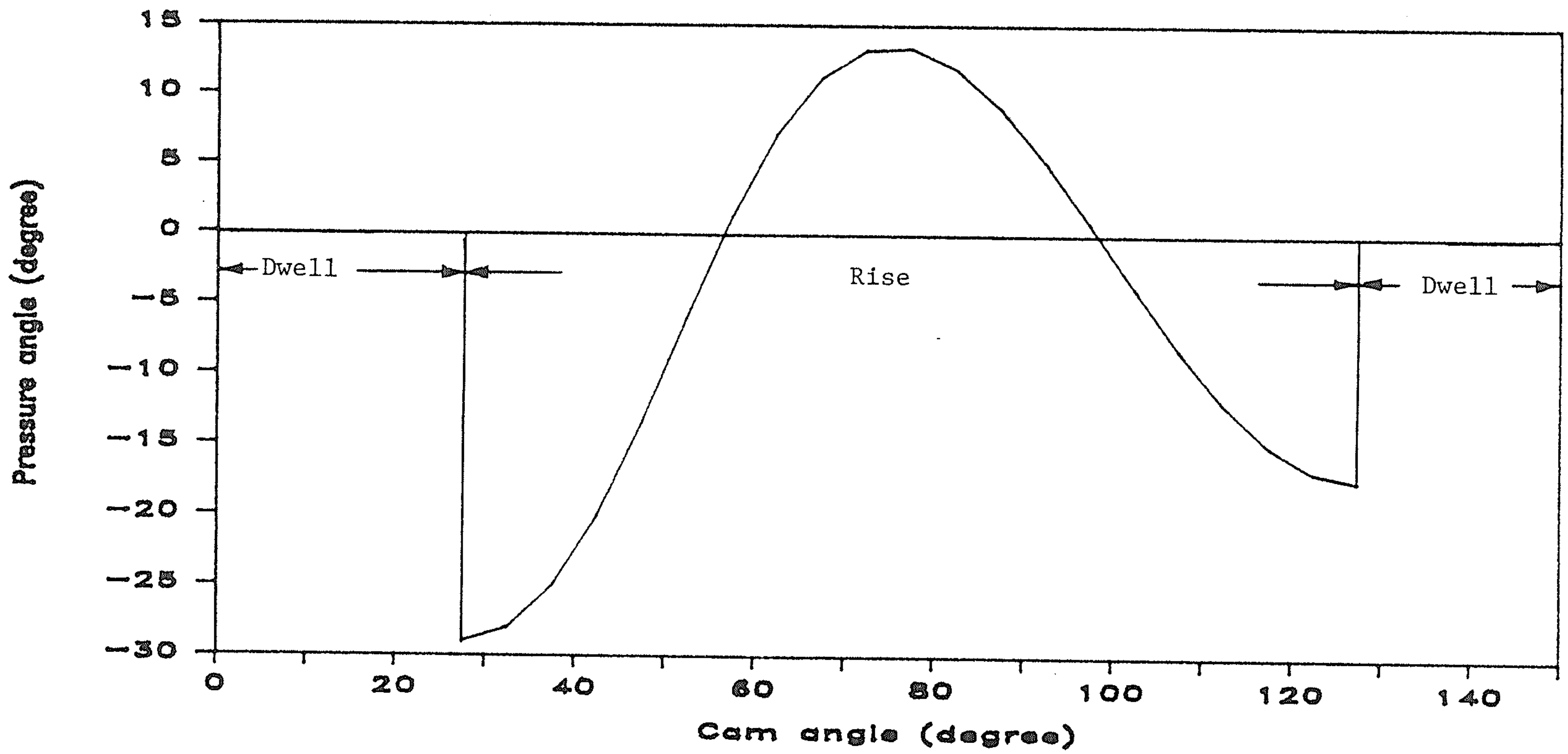


Figure 14: Pressure angle for different cam angles of cam used in redesign

3.10 UNDERCUTTING PHENOMENON AND ITS ROLE IN FOLLOWER MOTION

The cam size as determined above, is acceptable based on pressure angle considerations. Further investigation is necessary to determine if there is undercutting of the cam profile. Undercutting is the condition when the cam profile has an inadequate curvature so that correct follower motion is impossible. Undercutting is related to the roller radius and the minimum radius of the pitch curve of the cam profile.

The observed reaction depends on whether the cam is convex or concave. In the case of a concave cam the curvature of the cam profile is less than the radius of the roller follower The sharp point phenomenon and undercutting phenomenon will be involved simultaneously. In the case of a convex cam the sharp point phenomenon and the undercutting phenomenon will not occur simultaneously. If the radius of curvature of the pitch curve is less than the roller radius the undercutting condition will result but if the radii are equal, the sharp point phenomenon will occur. The sharp point phenomenon is nothing more than a condition where a stress concentration occurs. Wear is accelerated and therefore the condition should be avoided.

The two conditions can be expressed as follow. For a concave cam, sharp point or undercutting occurs if Rho < 0 ; for a convex cam, undercutting occurs if Rho < Rf and a sharp point occurs if Rho = Rf.

3.11 DETERMINATION OF UNDERCUTTING

Undercutting is related to the radius of curvature of both the roller and the cam profile. The minimum allowable radius of curvature of the pitch curve of a cam and the roller has to be determined. For the cam profile the radius of curvature varys except for the dwell portions. To determine the radius of curvature for the cam the general relation can be written as:

$$Rho = \pm A/B \qquad [3.11\text{-}1]$$

where,

$$A = [(R0+Y)^2+(Y')^2]^{1.5}$$

$$B = (R0+Y)^2+2(Y')^2-(R0+Y)(Y'')$$

The positive value is used with the convex cam and the negative value is used with the concave cam.

The radius of curvature is a function of the follower acceleration. From the general equation it can be seen that the maximum positive acceleration will result in the largest radius of curvature whereas the maximum negative acceleration will result in the least radius of curvature. Obviously the required acceleration determines the steepness of the cam. However, where these points are located depends mainly on the type of follower motion and the cam angle. In the case of simple harmonic motion the maximum positive and negative accelerations occur at the beginning and the end of the cam cycle. For cycloidal motion these points occur at the 1/4 and 3/4 points of the cam angle. In both cases the absolute magnitudes are the same. This is a disadvantage of pure trigonometric motion compared to constant acceleration motion.

With constant acceleration motion the two opposite accelerations can be made unequal. This is done by moving the inflection point away from the mid position of the stroke. Locating the inflection point ahead of the half stroke position will provide less negative acceleration and after the half stroke will result in less positive acceleration. The maximum negative acceleration can be made small and the minimum radius of curvature can be increased. This is a desirable characteristic of constant acceleration motion.

The acceleration for the cycloidal motion can be written as:

$$Y'' = 2pi(H)[\sin(2pi\ \phi/\beta)]/\beta^2 \qquad [3.11\text{-}2]$$

For a convex cam the minimum radius of curvature will occur when $\phi/\beta=0.75$. Therefore, the minimum radius of curvature of the cam profile will be Rho(min) and equation [3.11-2] becomes:

$$Y'' = -2pi(H)/\beta^2 \qquad [3.11\text{-}3]$$

$$Rho = Rho(min)\ (\text{at } \phi/\beta = 0.75)$$

where,

H = the total rise of the follower, mm

For a cam profile to avoid undercutting the following condition must exist:

$$Rho(min) > R(fol)$$

[3.11-4]

where,

Rho(min) = minimum radius of curvature of cam profile, mm

R(fol) = radius of curvature of roller follower, mm

If the condition is not satisfied then a reduction in roller radius or an increase in cam size is necessary. In the computer program (Appendix D) the roller size is reduced until the undercutting is eliminated. Kent (1958) suggests that the minimum radius of curvature of the cam profile at any point should not be less than d+3.2 mm (d = diameter of the roller, mm). Otherwise there will be a loss of rise distance for the follower which will vary depending on the sharpness of the angle of intersection of the two adjoining curves on the cam profile.

For the design of the width of the roller, Berard and Waters (1924) suggest that if the load on the cam is known the roller width may be computed on the basis of stresses of 3.44 MPa. In other cases the roller width can be calculated as equal to 1/4 (roller diameter)+3.53 mm (Berard, 1924). The load on the cam is assumed very small in this case. The width of the roller is not critical and the roller pin should be sized 1/3 to 1/2 the diameter of the roller.

3.12 DEVELOPMENT OF CAM PROFILE WITH ROLLER FOLLOWER

With the cam and the roller design parameters thoroughly investigated for problem free operation, it is now necessary to determine the cam profile. Referring to Figure 15, the development of the cam with optimal offset of a roller follower can be represented by the following general equation. The general equation for the cam profile (Chen, 1982):

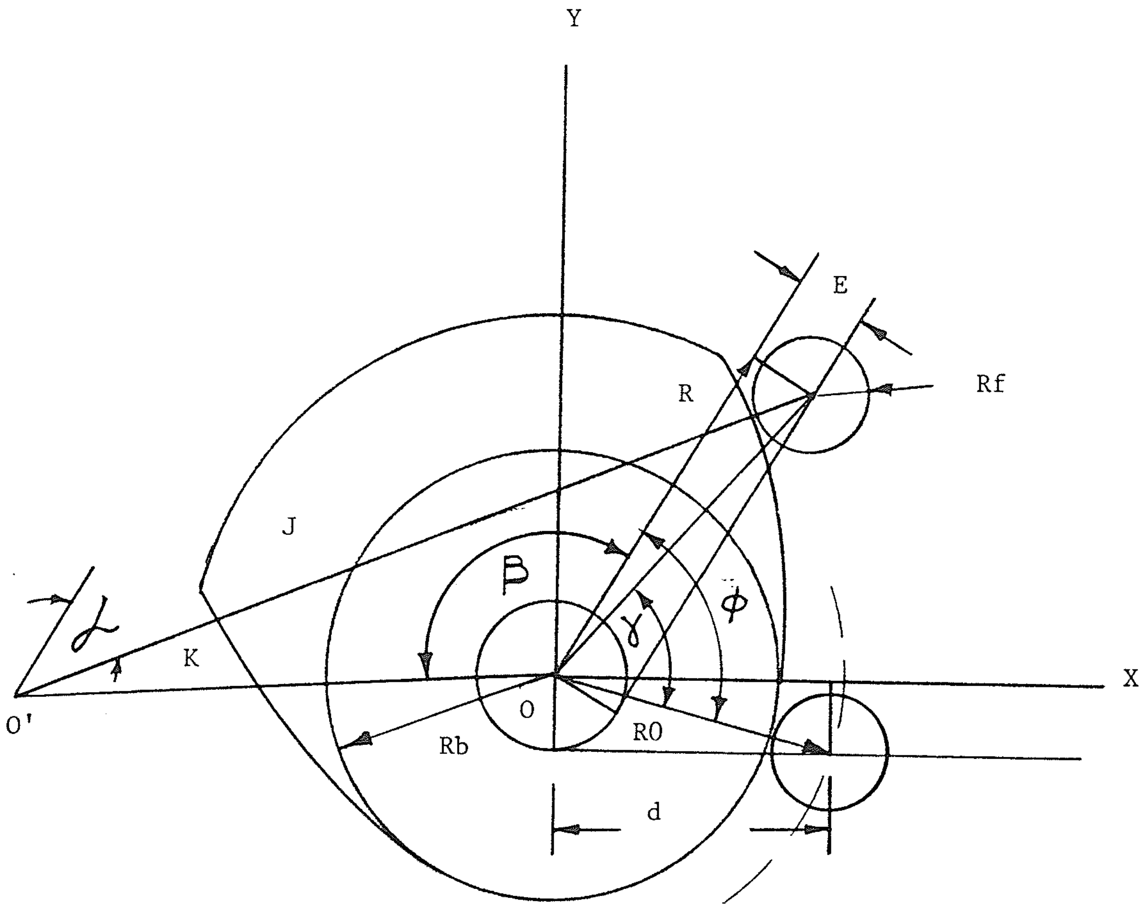


Figure 15: Determination of cam size with roller follower

$$F(X,Y,\gamma) = [X-M-(d+s)\cos\gamma]^2+([X+N-(d+s)\sin\gamma]^2-(Rf)^2$$

[3.12-1]

Where,

$d = [(Rb+Rf)^2-E^2]^{0.5}$, mm

Rb = Base radius of cam, mm

Rf = Radius of roller follower, mm

s = Displacement of the follower, mm

M = $E(\sin\gamma)$, mm

N = $E(\cos\gamma)$, mm

E = offset distance, mm

γ = angular displacement of cam from initial position to the point considered, rad.

G and H can be defined as follows:

$$G = (d+s)\sin\gamma - (E + ds/d\gamma)\cos\gamma$$

$$H = (d+s)\cos\gamma + (E + ds/d\gamma)\sin\gamma$$

Equation [3.12-1] is differentiated with respect to γ and set equal to zero to give the X and Y coordinates of the cam profile. The coordinates are:

$$X = E(\sin\gamma)+(d+s)\cos\gamma+(Rf)/[1+(G/H)]$$

[3.12-2]

$$Y = [X(G)+(d+s)(ds/d\gamma)]/H$$

3.13 DESIGN OF FOLLOWER SPRING

The purpose of the spring on the cam follower is to counteract the inertia of the follower and to keep the follower in contact with the cam at its maximum negative acceleration period. Another purpose is to store potential energy at the end of the lifting stroke. This energy is to be released when the engaging tooth reaches the predetermined position. The inertia force helps the spring to keep the follower on the cam when the cam follower has positive acceleration. However, when the follower has negative acceleration, the inertia force tends to lift the follower off the cam. A jump would occur when the negative inertia force of the system exceeds the available spring force and other counteracting forces.

The occurrence of negative acceleration (deceleration) does not depend so much on the follower rise or fall but rather on the type of follower motion used. For a follower with cycloidal motion, negative acceleration occurs even on the rise. The deceleration starts at the mid point of the total cam angle and attains its maximum magnitude at the three quarter point of the total cam angle. The magnitude is a function of the angular velocity of the cam, the total cam angle and the corresponding rise. The spring force, together with the external load, the weight of the follower and the frictional resistance force between the follower stem and the guides, must be at least equal to the inertia force due to the maximum deceleration so that the follower will always be in contact with the cam. A force balance on the follower at the moment of separation from the cam can be expressed as:

Inertial force = force due to follower weight

+ force due to friction in the guides

+ force due to spring

+ force due to any external load

In the above word equation the spring force is of interest. There will be no external load except when the engaging tooth hits a roller momentarily. It is not possible to calculate the external load exactly for this condition. The required spring stiffness has been estimated at 540 N/m (see Appendix G). Calculation of the spring stiffness may seem to be unnecessary. But, too much force will increase wear at the contact point of the cam and the follower. A larger spring force would also require more power. Another important consideration is the resulting dynamic load when the tooth is released by the trip mechanism. The spring force is also required for estimating the power needed for the trip mechanism. The power required has been estimated as 2 W approximately (see Appendix G).

3.14 SELECTION OF MATERIALS FOR THE FOLLOWER AND CAM

Material selection will be based on service life of the mechanism and economy. The cam and the follower could fail for two reasons. One reason is due to breakage of the cam or follower. This would only occur when unexpected large external loads were applied to the system. Since the components are designed with enough strength for ordinary circumstances, breakage will seldom occur. Therefore breakage will not be considered.

The other reason for failure is excessive wear. The term wear may be described as a progressive loss of surface material by shearing and tearing away of particles. According to Almen (1950), wear can be classified as:

1. Removal of metal associated with (a) corrosion, (b) abrasion and (c) fatigue or pitting.
2. Transfer of metal between surfaces due to contacting macro-points under high temperature while sliding.
3. Displacement of metal by plastic flow, superficial wear or smearing. This occurs under high pressure.

In this design the type of wear is most likely concerned with the removal and transfer of metal between surfaces. Also, since the working conditions of the machines are wet and muddy, corrosion and abrasion could be problems. Machine components exposed to a corrosive atmosphere will have a lowered fatigue resistance. This is because of roughing or pitting of the surface by corrosion. To avoid corrosion it is necessary to choose corrosion resistant material. Proper cleaning and the application of grease on the appropriate parts, before and after use, could minimize the problem of corrosion.

Abrasion involves the presence of some foreign material between the contact surfaces. The probable foreign material in this case is sand or dirt. This problem seems to be unimportant in the case of a roller follower. The system loads are small and at the same time most of the intruding foreign material would probably be softer than the cam or follower material. However, in the case of a flat-faced follower foreign material could be serious because of galling action. This particular problem would not be expected in the case of a roller follower.

Transfer of material by a combination of welding, scoring, galling, and scuffing is a special problem which is more likely to occur with a flat-faced follower. Scoring is due to the action of pressure, sliding velocity, and the resulting high instantaneous temperatures. However, this problem could also occur with a roller follower if the roller does not have pure rolling action. Most roller followers exhibit some sliding even at low speeds. Therefore the problem needs to be considered even in the case of a roller follower.

Fatigue wear begins with a microscopic crack at a critical area of high local stress or at a local discontinuity such as a notch. The application of repeated loading beyond the local plastic limit is the usual cause of fatigue failures. Careful analysis reveals that the actual maximum average stresses were below the ultimate strength of the material and quite frequently even below the yield strength. Stresses repeated a very large number of times are the main cause of failure which is usually called fatigue. Fatigue failure occurs with no warning in advance. It is sudden and dangerous.

The understanding of surface fatigue is not quite complete at this time. Some current points that are known are:

1. Increased surface hardness increases resistance to surface fatigue.
2. The depth of surface hardness should be twice the distance over which maximum shear stress is believed to act.

3. Close-grain structure gives good resistance to fatigue.

4. Appropriate shot peening, cold working and better surface finishing improve the endurance limit and consequently increase the life of a cam.

5. Reduction of the applied loads and the sliding velocity will minimize surface fatigue.

3.15 STRESS DEVELOPMENT BETWEEN CAM AND FOLLOWER

When two surfaces roll against one another with sufficient force a pitting failure will occur after a certain number of cycles of operation. In the case of a roller follower wear is associated with contact stresses developed between the contact surfaces of the cam and the follower. In a Hertzian rolling contact it has been established that a high shear stress zone exists at a depth below the rolling surface and that surface spalling frequently initiates in this subsurface high shear stress zone. Lundberg and Palmgren reported that most failures start at the depth of maximum reversing shear stress. Metallographic observations made by Martin (1967) revealed that plastic deformation bands are found very close to the rolling surface at locations which have undergone surface fatigue. In support of this, Chen (1982), reported that surface fatigue is the dominant cause of failure in cam and follower systems.

The actual contact stress is influenced by the applied loads, the geometrical shape of the contacting surfaces and the material properties. The geometrical shape of the surface is dependent upon the radii of the curvature of the two bodies and their alignment. It is difficult to relate these parameters in mathematical terms. Some simplifying assumptions need to be made. Test results in laboratory tests do not predict accurately results of actual use. Therefore, much on the subject of surfaces in contact is controversial and experts disagree as to the phenomena of surface stress and life expectancy. At present there are no fully acceptable empirical formulae for predicting life. Consequently, most design books state that the theory of contact stress and deformation is one of the more diffcult topics in the theory of elasticity.

The best known relationship was introduced by Hertz. Hertz theory is based on the assumptions that the contacting bodies are perfectly elastic, that there are no shearing stresses in the contact area, and that the radii of curvature are large in comparison to the dimensions of the area of contact. As a result of Poisson's ratio the stresses developed will be on a load axis as well as on the lateral axis.

In the case of two cylinders the stresses along the load axis and longitudinal axis of the cylinder will be almost the same. However, on the transverse axis the stress developed will be about one half of the amount of the other two directions. The stresses are all compressive because of the small contact area compared to the radii of the two cylinders. The stresses are diminished with depth. At a certain depth, about one half of the length of the contact surface in the transverse plane, maximum shear will be developed. This is the possible origin of fatigue failure.

Hertz established contact stresses in relation to these parameters for different configurations of contacting bodies such as two spheres, two cylinders, or a sphere and a cylinder, etc. Hertz also gives the contact stresses between two cylinders which is the case of the cam and the follower.

Mathematically, these stresses are (Juvinall, 1983):

$$P_0 = 0.564\ (AA/BB)^{0.5} \qquad [3.15\text{-}1]$$

where,

$AA = [(F/W)(R1+R2)]/[(R1)(R2)]$

$BB = [(1-\delta 1^2)/E1] + [(1-\delta 2^2)/E2]$

P_0 = maximum stress, MPa

R1 = radius of curvature of roller, mm

R2 = radius of curvature of cam, mm

E1,E2 = the moduli of elasticity of the cam and follower, MPa

F = the applied force, N

$\delta 1$ = Poisson's ratio of roller follower, unitless

$\delta 2$ = Poisson's ratio of the cam, unitless

L = width of the cam or roller whichever is small, mm

or width of the contact surface.

In the case of two similar metals the equation reduces to:

$$P_0 = 0.418[E(F/L)(R1+R2)/\{(R1)(R2)\}]^{0.5} \qquad [3.15\text{-}2]$$

For the flat-faced follower, R1 is infinite and equation [3.15-1] becomes,

$$P_0 = 0.54[F/\{(L)(R2)(BB)\}]^{0.5}$$

[3.15-3]

Similarly, for two identical metals,

$$P_0 = 0.418[\{(F)(E)\}/\{(L)(R2)\}]^{0.5}$$

[3.15-4]

3.16 LIFE EXPECTANCY OF THE CAM MECHANISM

In the case of a roller follower it has been seen that wear is a function of the contact compressive stresses. If the compressive stresses developed are large wear will be excessive. If the stresses developed are small then the wear life will be long. Way (1935) reported that pits were found to have a definite characteristic shape and orientation with respect to the direction of the rolling action. The pits developed from minute cracks in the roller surfaces after about 500,000 revolutions. He did not mention the amount of load applied.

Buckingham and Talbourdet ran tests on cylindrical roller followers in contact with a cam surface under controlled conditions of rolling and alignment. They used different materials with various hardnesses and surfaces in contact. They found that the allowable stresses for the combinations shown in Table 6 were greater than the stresses computed using the Hertz formula. Hertz stresses are indicative of a life of more than $10X10^6$ cycles in a system with pure rolling. They further found that the above holds true even in the case where sliding is evident in

TABLE 6

Properties of cam materials

Cam material for use with roller of hardened steel	Maximum allowable compressive stress MPa
Gray-iron casting, ASTM 48 - 48, class 20, 160 - 190 BHN phosphate coated	379
Gray-iron casting, ASTM A 339 - 51 T, grade 20, 140 - 160 BHN	351
Nodular-iron casting, ASTM A, 339 - 51 T, grade 80 - 60 - 03, 207 - 241 BHN	496
Gray-iron casting, ASTM A 48-48, class 30, 200 - 220 BHN	448
Gray-iron casting, ASTM A 48 - 48, class 35, 225 -255 BHN	538
Gray-iron casting, ASTM A 48 - 48, class 30, heat treated (Austempered), 225 - 300 BHN	620
SAE 1020 steel, 130 -150 BHN	565
SAE 4150 steel, heat treated to 270 - 300 BHN, phosphate coated	1517
SAE 4150 steel, heat treated 270 - 300 BHN	1296
SAE 1020 steel, carburized to 0.045 in depth of case, 50 - 58 RC	1558
SAE 1340 steel, induction hardened to 45 - 55 RC	1365
SAE 4340 steel, induction hardened to 50 - 55 RC	1558

Source: (Erik Oberg, 1984. Machinery's handbook) the system. With sliding a certain modifying factor must be introduced into the formula. The use of the factors is illustrated as follows:

Case 1. (sliding of 0 - 10 percent)

4/5 [allowable stress from Table 6] > stress (computed)

Case 2. (sliding of 11 - 75 percent)

1/3 [allowable stress from Table 6] > stress (computed)

The amount of sliding must be estimated from analysis of the system under study. The friction moment tending to rotate the cam and the resisting moment can be expressed as:

$$Mc = (F)(Rf)(U) \tag{3.16-1}$$

$$Mp = (F)(r)(U1) \tag{3.16-2}$$

where,

Mc = moment for rotation, N-m

Mp = resisting moment, N-m

F = load developed in the system, N

Rf = radius of the roller, m

r = radius of the roller pin, m

U = friction coefficient between roller and cam

$U1$ = friction coefficient between roller and pin

Since plain journal bearings are assumed, the difference in the two friction values will not be significant compared to their radii of application. Therefore, the moment at the cam surface will be greater than the moment at the roller pin unless or until the bearing surface is dirty and jammed. Therefore, sliding will not likely occur at the cam surface. It has been calculated that this frictional resistance is large enough to overcome starting inertia. It can be deduced that sliding at roller surface cannot exceed 49 percent. Buckingham's finding may be applied in this case. Any combination of sliding having allowable stresses no greater than three times the computed stresses using the Hertz formula will last at least $10X10^6$ cycles which is equivalent to more than 3000 hectares.

In the case of a flat-faced follower the contact stresses developed at the contact surface will be less than those developed in the case of a roller follower. However, it cannot be concluded that the service life will be as above since the sliding percentage is now more than 75 percent. Buckingham's findings are no longer valid. The case must be treated as two rubbing surfaces. Wear rates will depend on the material surface roughness and hardness. Wear will also depend on the dynamic loads and the velocity of the two bodies. The original surface finish will be another factor.

Wear phenomena is very complicated and therefore experimental results must be considered. Rothbart (1956) compiled some imformation about wear. Some of the information is related to this topic and listed as follows:

1. Gray cast iron in combination with hardened steel is probably the cheapest choice where high sliding and galling action may occur.

2. Chilled cast iron and alloyed cast iron are among the best materials.
3. Bronze or nylon in combination with hardened steel has good wear life, less noise, less vibration, and smoother action. These materials also compensate for inaccuracies in the contour surface especially under high shock loads.
4. For high sliding conditions, Meehanite in contact with hardened steel is an excellent choice giving long life, good internal vibratory damping characteristics and easy machining.
5. Hardened tool steel in contact with hardened steel gives the best combination under rolling action when shock and sliding action occur.
6. Good surface finish with lapping and polishing is essential with hardened materials. It is not so important with ductile materials.

By studying this and other information one general corollary can be drawn. Similar metals in contact do not give good wear resistance. This does not mean that all dissimilar metal combinations give good wear resistance. The proper choice of metal combination is still necessary. Most literature leaves the selection entirely to the experience of the designer because the wear problem is very complicated.

Based on the situation of the design, the availability, cost and wear life of materials, a decision has to be made. From the stand point of economy and easy availability the follower surface (or the roller) can be made of hardened and ground steel (Rockwell C 51 - 52). The cam surface material should be either gray cast iron or chilled cast iron whichever is cheapest or more readily available. If neither of these are available then ether cast iron or steel may do. If either of the follower or flat-faced follower must be made of mild steel it is recommended that the cam be made of harder material since it is easier to replace the followers than the cam.

For flat-faced followers with pure sliding the wear action will involve galling. Analytical approaches to the wear of two rubbing surfaces were developed in the 1940's. One such development is as follows (Juvinall, 1983):

$$W = (K/H)(F)(S) \tag{3.16-5}$$

where,

W = volume of material worn away, mm^3

F = compressive force between the surfaces, N

S = total rubbing distance, mm

K = wear coefficient, dimensionless

H = surface hardness, MPa (Brinnel hardness number X 9.81)

The wear coefficient is highly influenced by lubrication. The wear coefficient can be reduced by a factor of 1/10000 by proper lubrication. Wear coefficients are mutual dependent. Generally, the wear coefficients are greater with similar metals. For compatible metals the wear coefficient in a lubricated condition is approximately 1/1000 of the unlubricated case (Juvinall, 1983). For a total compressive force between the surfaces, the volume of material worn away is independent of the area of contact. Applying these criteria to this situation the number of cycles necessary to wear away 0.01 mm of the surface will be $23X10^3$ cycles, equivalent to 7 hectares.

Chapter IV

OPERATION AND EVALUATION

4.1 KINEMATICS OF THE INDEXING MECHANISM

The trip mechanism is activated by the cam mechanism. The cam oscillates whenever the operator pushes and pulls the handle. The cam action is effective only when the follower is exactly above the cam and only when the operator executes the pull stroke. As the operator pulls the handle the cam turns 144 degrees. The cam angle of 144 degees is divided so that the first 27 degrees is a dwell with the next 100 degrees producing a 14 mm rise with cycloidal motion followed by approximately 17 degrees of dwell.

The cam angular velocity is zero initially. Due to the nature of the operator's activation of the mechanism, the angular velocity of the cam over the working or lifting sector is approximately constant. The follower is lifted against the compression spring with finite jerk at a maximum of 13 m/s^3 based on the design for cycloidal motion. A collar with a groove is attached to the follower stem and moves upward with the follower (see Figure 16). A holding device under spring tension allows the collar to move upward only. The holding device does not allow the downward movement of the collar unless an external load is applied. Under the above action the upper tooth is disengaged from the chain and the lower tooth is engaged.

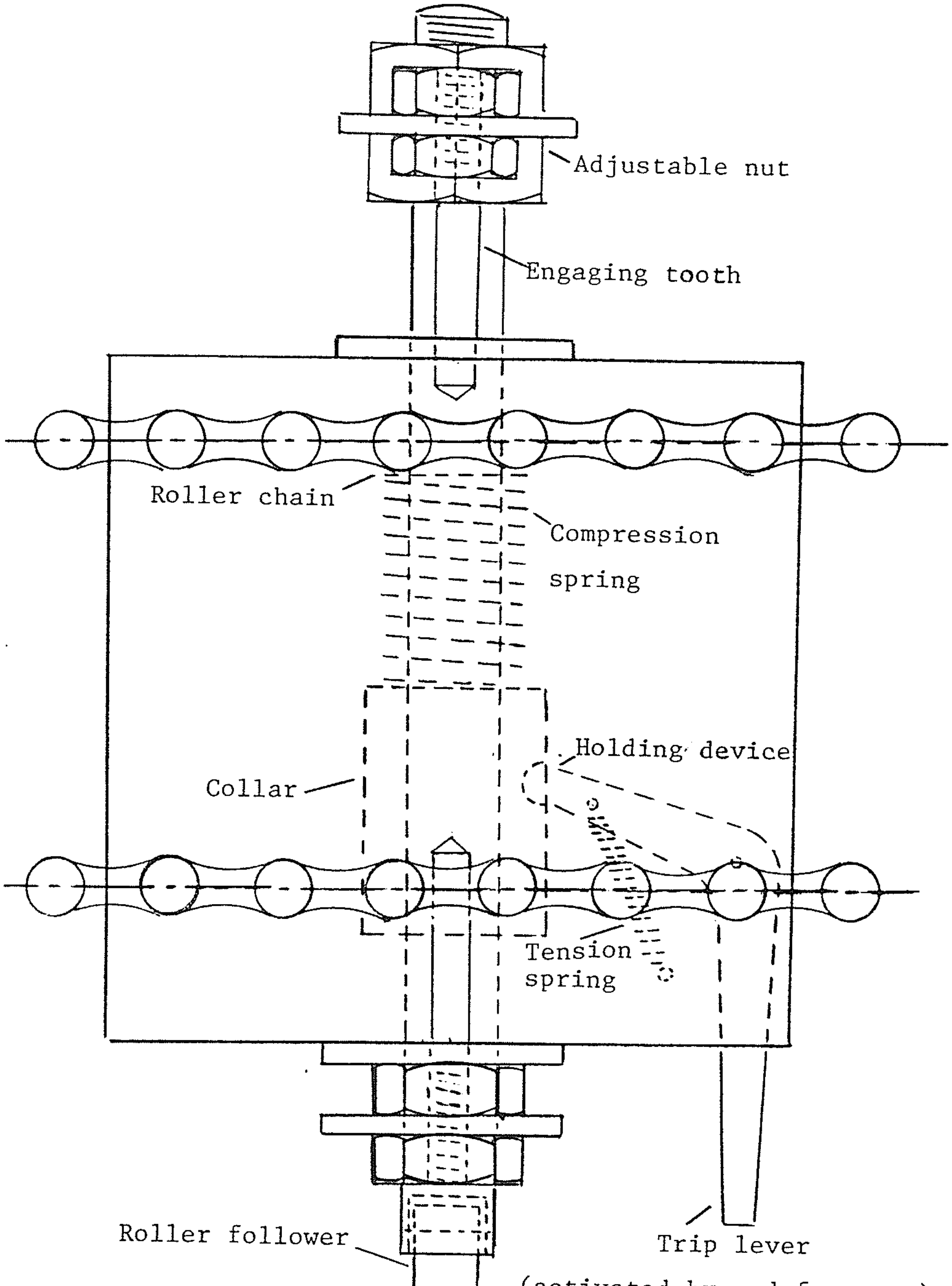


Figure 16: Schematic diagram of trip mechanism

Further action of the handle moves the whole unit to the next planting position. The operator pushes the handle turning the cam backwards. The follower and the lower tooth which is engaged are under force to reset by the action of the compression spring. But the holding device locked into the groove on the collar holds the follower so that the lower tooth remains in the engaged position.

The action of the picker arm and the tray drive lever can be explained as follows. The picker arm which is part of the input link of the four bar linkage turns toward the operator to its final position (74 degrees from the horizontal) (see Figure 6). When it has returned 36 degrees attaining a position of 38 degrees from the horizontal, the tray drive lever activator which is also attached to the picker arm contacts the tray drive lever. This causes the drive sprocket to turn through a certain arc giving linear movement to the chain. The engaging tooth inserted into the chain moves with the chain. The transplanter tray which is fastened to the engaging tooth moves also. This sequence of actions occurs with every stroke by the operator.

The tray finally reaches an extreme position which is approximately 200 mm from the first disk cam. Another cam, an end face type is engaged and activates the trip mechanism. The angular movement of the end face cam is similar to the disk cam since they are both mounted on the same shaft. The function of the end face cam is to reverse the tray movement. The follower holding device is released by the axial movement against the end face cam. This permits the compression spring to release its energy pushing the follower down. The downward movement of the follower engages the upper tooth in the chain and disengages the lower tooth. The operator inputs another stroke and the tray reverses. Thus, the reciprocating movement of the tray is achieved.

4.2 TESTING WITH SEEDLINGS

Wheat seedlings were prepared according to the instructions in the transplanter operator's manual. Test runs were carried out in the workshop. Due to the weak characteristics of the wheat shoots the picker did not pick the seedlings cleanly. Other attempts were made with oat and rye seedlings but these were even more fragile than the wheat seedlings. Trials with seedlings were abandoned. However, test runs without seedlings were carried out. This was justified since the modified part of the machine was not directly concerned with the seedlings. A test of 100 strokes was made. The tray movement was consistent with no observable missed index movements.

4.3 DISCUSSION

The main objective of the modification was to eliminate the dead spots and unequal movement of the indexing mechanism. A secondary objective was to reduce backlash and to improve the rigidity and durability of the mechanisms. In the modified design, no dead spots or unequal lateral tray movements were observed. The main objective has been fulfilled.

The cotter pin which was used as a connector and the entire indexing mechanism in the existing machine were replaced with the cam and trip mechanism so that the secondary objective can also be considered to be achieved.

However, there are minor problems with the modified mechanism. Unavoidable backlash between the follower guide and the follower stem exists. More precise fabrication could reduce this problem but it cannot be eliminated. Nevertheless, jamming was not a problem. Jamming was avoided by making generous allowances for the effect of friction in the system. For example, the maximum allowable pressure angle was calculated by assuming 1.5 times the expected static friction as most experts suggest for this situation. Moreover, the working pressure angle was limited to 50 percent of the computed maximum allowable pressure angle. In other words, the safety factor for the maximum allowable pressure angle was approximately two. Conditions were further improved by using an initially compressed spring on the follower which minimizes backlash in the system.

Other advantages of the modified mechanism over the existing machine are that any convenient chain sprocked size can be used and the length of the cycle of tray travel can be varied as desired. The engaging teeth are also easily replaceable. The depth of insertion into the chain links can be adjusted. The depth of insertion is related to cam movement and so unnecessary adjustment should be avoided. The original penetration was set to 14 mm only.

The follower could possibly misalign on the disk cam. This would be possible if the cam width were less than the maximum lateral movement of the tray during one stroke. If misalignment did occur, the tray would not change direction. To aviod this problem, a cam with a width of more than the maximum possible movement of the tray per stroke was used.

In the new design of the indexing mechanism both a flat-faced or a roller follower are possible. If the flat-faced type is used the cam size can be smaller. Another advantage of a flat-faced follower is that the pressure angle is constant and almost zero. But any changes in cam size will require a corresponding change in the overhang length of the follower and in the cam shaft location. The different overhang length will also change the maximum allowable working pressure angle. The change in the location of the cam shaft will also change the output angle of the cam.

The design was finalized after careful consideration of all of these factors. In the final design the cam size for a flat-faced follower was increased from the computed minimum size and made equal to the size obtained in the case of a roller follower. The vertical locations of both the follower and the cam shaft were made adjustable. The required location of the cam shaft for a flat-faced or a roller follower was made identical. Therefore, the use of a different follower either flat-faced or roller follower is possible without changing the cam shaft location. Note that the roller diameter cannot be changed.

Slight changes in the location of the cam shaft are very important. Changes in the vertical location will result in deficient or excessive lift of the follower. Changes in location of the cam shaft in a horizontal direction will effect the offset distance. Any offset distance, other than the optimal, will increase the pressure angle which could jam the trip mechanism. Therefore, accurate location of the cam shaft is important in the design. Similar problems can be expected with too much deflection of the cam shaft. Use of a more rigid cam shaft is recommended. Since the cam is of the dwell-rise-dwell type it can be made symmetrical and both faces used. In calculating the spring stiffness the follower weight is controlled by the weight of the collar. A low collar weight will give for better performance.

Chapter V

CONCLUSIONS

The indexing mechanism of an existing machine was replaced with a four bar linkage, cam and trip mechanism. By eliminating potentially missed hills this design improves planting performance by a minimum of 5 percent. The overall improvement due to greater uniformity of plant population and saved labor for filling missed hills is even more significant.

The design calculations were done by a computer program called KI-TRAN. Based on the calculations the following conclusions were drawn:

1. The four bar mechanism has a driver link of 117 mm, a coupler link of 432.22 mm, a base link of 515.3 mm, and a driven link of 35 mm.
2. The trip mechanism has an overhang length of 75 mm and a guide length of 55 mm and was selected for a maximum pressure angle of 60 degrees with a roller follower.
3. The cam mechanism with either flat-faced or roller follower has cycloidal motion with finite jerk throughout the motion events.
4. The cam imparts motion of a dwell-rise-dwell type. The first 27 degrees are dwell and the next 100 degrees have a rise of 14 mm followed by 17 degrees of dwell. The return stroke is similar.
5. The width of the cam face was 10 mm with an offset distance of 10 mm.

6. The width of the flat-faced follower was 52 mm with a cam size of 20.7 mm for base radius.
7. The roller follower roller was 8 mm in diameter with the above cam size (pitch radius = 20.7 mm).
8. The flat-faced or roller follower should be made of ground steel (Rockwell C 51 - 52) and the cam should be made of gray cast iron.

Chapter VI

RECOMMENDATIONS

All components are interrelated to each other for proper kinematic function and therefore any changes in size or location will create many problems. For the roller follower, offseting is very critical and even a slight deviation will cause a variation in pressure angle. To minimize the above considerations and to ensure proper machine operation it is recommended that:

1. A flat-faced follower be used.
2. The machine parts be made as rigid as possible with due consideration to keeping the total mass as low as possible.
3. All moving parts be as light as possible and that all acceleration be as low as possible.
4. All backlash be held to a minimum.
5. Minimum overhang length and maximum bearing guide length be used in order to limit the actual pressure angle.
6. Operators receive proper instruction as to using full strokes in order to avoid any malfunction of the trip mechanism.
7. Improvements in the holding device on the trip mechanism be considered.

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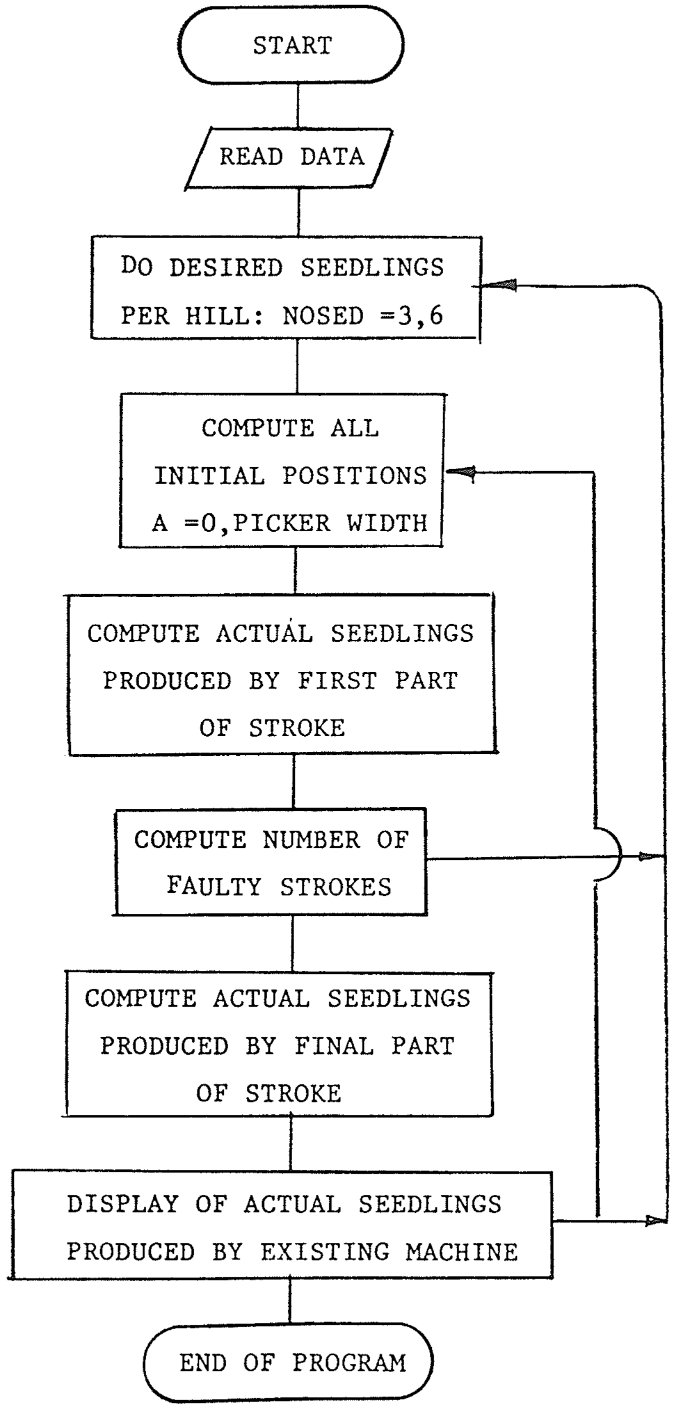
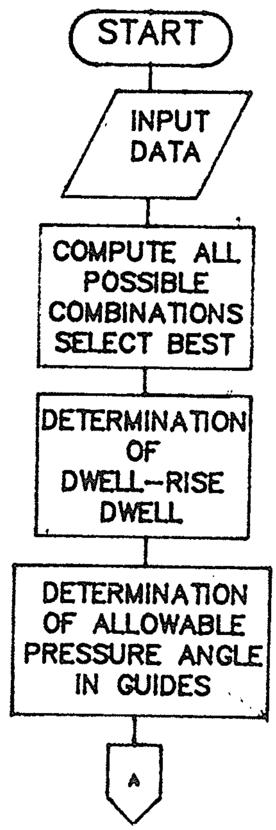
Appendix A

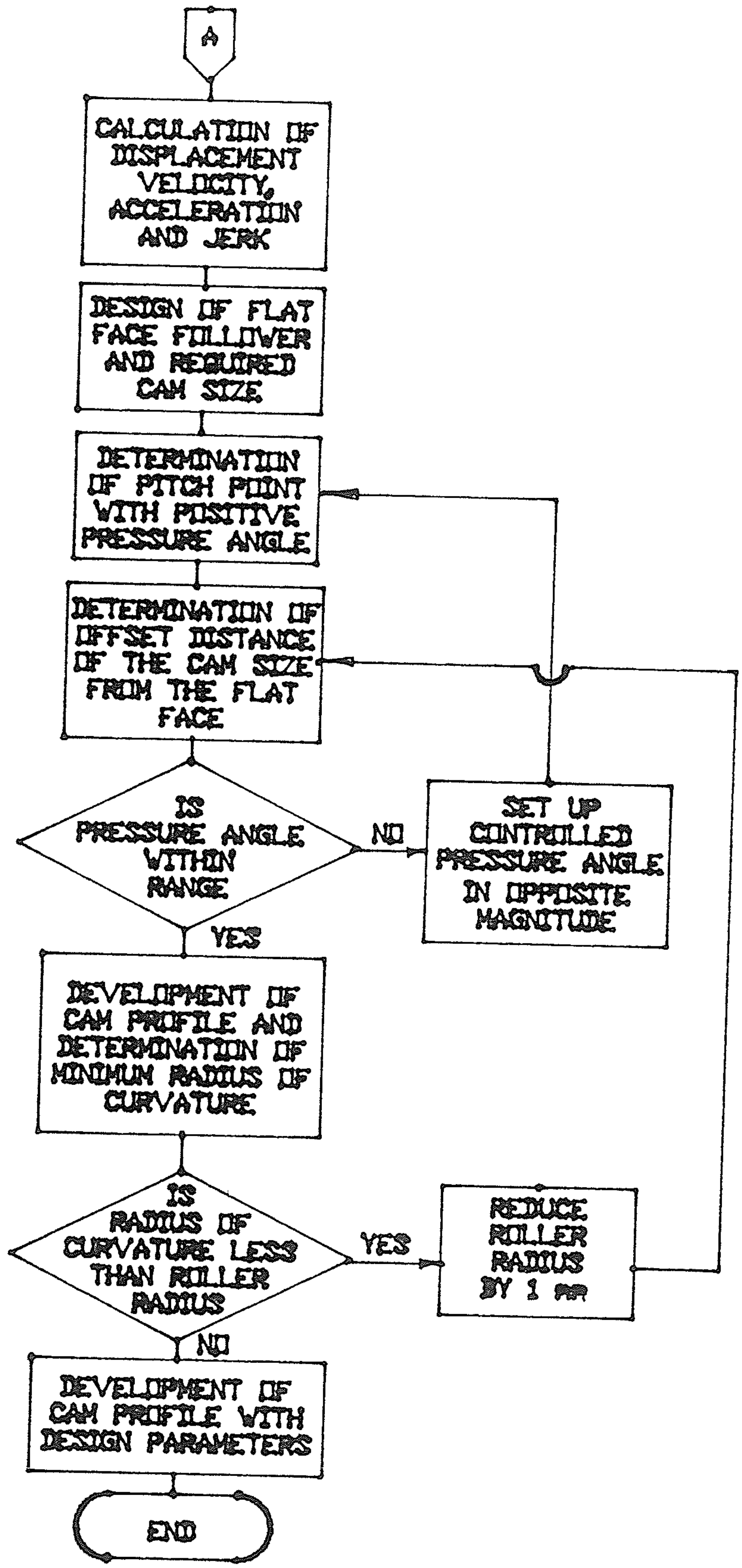
FLOWCHART FOR SE-LEFT



Appendix B

FLOWCHART
FOR
KI-TRAN




A
CALCULATION OF DISPLACEMENT VELOCITY, ACCELERATION AND JERK
DESIGN OF FLAT FACE FOLLOWER AND REQUIRED CAM SIZE
DETERMINATION OF PITCH POINT WITH POSITIVE PRESSURE ANGLE
DETERMINATION OF OFFSET DISTANCE OF THE CAM SIZE FROM THE FLAT FACE
IS PRESSURE ANGLE WITHIN RANGE
NO
SET UP CONTROLLED PRESSURE ANGLE IN OPPOSITE MAGNITUDE
YES
DEVELOPMENT OF CAM PROFILE AND DETERMINATION OF MINIMUM RADIUS OF CURVATURE
IS RADIUS OF CURVATURE LESS THAN ROLLER RADIUS
YES
REDUCE ROLLER RADIUS BY 1 mm
NO
DEVELOPMENT OF CAM PROFILE WITH DESIGN PARAMETERS
END

Appendix C

PROGRAM SE-LEFT

```
C  TINT JOB '022716,,,T=20,I=10,L=10,','TINT',NOTIFY=TINT,
C  PASSWORD=
C  EXEC WATFIV,SIZE=768K
C  SYSIN DD *
C  JOB  WATFIV   TINT,NOEXT
C  PROGRAM                 ***** SE-LEFT *****
                          $$$$$$$$$$$$$$$$$$$$$$$$
C  PROGRAMMER     ****** MAUNG SOE TINT ******
C                        $$$$$$$$$$$$$$$$$$$$$$$$$
C  PURPOSE : TO FIND THE DEFICIT IN EFFICIENCY OF INDEXING
C            MECHANISM OF RICE TRANSPLANTER TR1 SUCH AS :
C            NUMBER OF MISS HILL AND ACTUAL SEEDLINGS
C            PRODUCED PER HILL BY FAULTY STROKES.
C                        $$$$$$$$$$$$$$$$$$$$$$$$
      INTEGER N,T,ARRAY(10),LAPIN(10),NOSED,ISELEF,FSELEF,
     *COL(20),MAT(10,20),I,K,LMOVV,Z
      REAL RRAD,LMOV,AA,BB,THETA,THETA1,THETA2,X,LMOVF,
     *LMOVIN,XINTIL,XFINAL
C       *****   VARIABLE DICTIONARY   *****
C
C
C N         ... NUMBER OF STROKES REQUIRED TO COMPLETE THE
C               DISTANCE BEYOND ROLLER CENTRE
C T         ... LOOP COUNTER
C ARRAY(T)  ... SINGLE MATRIX TO STORE LENGTH OF LINEAR
C               MOVEMENT CORRESPONDING TO EACH STROKE BEYOND
C               CENTRE OF ROLLER
C LAPIN(T)  ... SINGLE MATRIX TO STORE THE ACTUAL SEEDLINGS
C               PLANTED WHEN CONNECTING JOINT IS BEYOND THE
C               POINT OF THE ROLLER
C LMOV      ... LINEAR MOVEMENT OF TRAY PER STROKE
C AA        ... HORIZONTAL DISTANCE BETWEEN CENTRE OF ROLLER
C               AND LAST POSITION OF COTTER PIN BEFORE PASSING
C               ROLLER CENTRE
C BB        ... HORIZONTAL DISTANCE BETWEEN CENTRE OF ROLLER
C               AND LAST POSITION OF COTTER PIN AFTER PASSING
C               ROLLER CENTRE
C THETA1    ... PART OF ANGLE PRODUCED BY PART OF STROKE BEYOND
C               THE CENTRE OF ROLLER
C THETA2    ... PART OF ANGLE PRODUCED BY PART OF STROKE BEFORE
C               THE CENTRE OF ROLLER
C THETA     ... ANGLE PRODUCED BY ONE COMPLETE STROKE BEYOND
C               CENTRE
C XINITL    ... HORIZONTAL LENGTH OF MOVEMENT PDODUCED BY PART
C               OF STROKE AT BEGINING
```

```
C XFINAL    ... HORIZONTAL LENGTH OF MOVEMENT PRODUCED BY PART
C               OF STROKE AT ENDING
C LMOVF     ... LINEAR MOVEMENT OF TRAY PRODUCED BY FINAL PART
C               OF STROKE
C LMOVIN    ... LINEAR MOVEMENT OF TRAY PRODUCED BY INITIAL PART
                OF STROKE
C ISELEF    ... SEEDLINGS PRODUCED AT FIRST HILL
C FSELEF    ... SEEDLINGS PRODUCED AT LAST HILL
C RRAD      ... RADIUS OF ROLLER IN MM
C NOSED     ... PREDICTED NUMBER OF SEEDS PER HILL
      READ,RRAD,LMOV,NOSED
      PRINT 10
      CARD(1)='1st'
      CARD(2)='2nd'
      CARD(3)='3rd'
      CARD(4)='4th'
      CARD(5)='5th'
   10 FORMAT('1',//T25,'ACTUAL NUMBER OF SEEDLINGS PRODUCED BY'/
     *       T29,'THE ORIGINAL INDEXING MECHANISM'/)
      THETA=LMOV/RRAD
      Z=1
      DO 999 NOSED=3,6
      PRINT 20,Z,NOSED
   20 FORMAT('0',//////////,T40,'TABLE',2X,I1,//////
     *      T25,'Predicted number of seedlings pre hill'//
     *      T25,'as connecting pin traverses end roller'//
     *      T23,'(desired number of seedlings = ',I1,1X,'per',1X,
     *      'hill)'//
     *      T29,'(nominal index length = 9.5 mm)'/)
      LMOVV=LMOV
      DO 30 T=1,LMOVV
   30 COL(T)=T-1
      PRINT 35
   35 FORMAT(/T14,'-----------------------------------',
     *      '--------------------------'//
     *      T29,'Connecting pin relative approach position (mm)'//
     *      T31,'(measured from the end roller centre line)'//)
      PRINT 40,(COL(T),T=1,LMOVV)
   40 FORMAT(/T14,'No. of stroke',3X,11(I2,3X))
      PRINT 45
   45 FORMAT('0',T14,'------------------------------------------',
     *      '---------------------')
      I=0
      AA=0.0
      WHILE (AA .LT. LMOV) DO
      I=I+1
      THETA1=(LMOV-AA)/LMOV*THETA
      XINITL=RRAD*COS(1.5708-THETA1)
      LMOVIN=LMOV-AA
      IF (LMOVIN .EQ.0.0) THEN DO
         GO TO 333
      END IF
      ISELEF=NOSED-ABS(LMOVIN-XINITL)/LMOVIN*NOSED+0.4
      MAT(1,I)=ISELEF
      N=0
```

```
      X=RRAD
      WHILE (X .GT. 0.0) DO
         X=RRAD*COS(1.5708-THETA1-N*THETA)
         N=N+1
      END WHILE
      N=N-2
      X=RRAD*COS(1.5708-THETA1-N*THETA)
      THETA2=3.1416-N*THETA-THETA1
      BB=LMOV-THETA2*LMOV/THETA
      LMOVF=LMOV-BB
      XFINAL=RRAD*COS(1.5708-THETA1-N*THETA)
      FSELEF=NOSED-ABS(LMOVF-XFINAL)/LMOVF*NOSED+0.4
      MAT(N+2,I)=FSELEF
      DO 100 T=1,N
         X=RRAD*COS(1.5708-THETA1-(T-1)*THETA)
         ARRAY(T)=ABS(RRAD*COS(1.5708-THETA1-T*THETA)-X)
         LAPIN(T)=NOSED-(LMOV-ARRAY(T))/LMOV*NOSED+0.4
         MAT(T+1,I)=LAPIN(T)
  100 CONTINUE
      AA=AA+1.0
      END WHILE
  333 CONTINUE
      K=N+1
      DO 90 T=1,K
   90 PRINT 95,CARD(T),(MAT(T,I),I=1,LMOVV)
   95 FORMAT(///T23,A3,5X,16(I1,4X)/////)
      PRINT 105
  105 FORMAT('0',//T14,'-------------------------------',
     *             '--------------------------------'////)
      Z=Z+1
  999 CONTINUE
      PRINT 1000
 1000 FORMAT('1'//)
      STOP
      END
$ENTRY
12.5,9.5,3
```

Appendix D

PROGRAM KI-TRAN

```
C  TINT JOB '022716,,,T=25,I=1,L=10,','TINT',NOTIFY=TINT,
C  PASSWORD=
C  EXEC WATFIV,SIZE=768K
C  SYSIN DD *
C  $JOB  WATFIV   TINT,NOEXT
C  PROGRAM                ***** KI_TRAM *****
C                      $$$$$$$$$$$$$$$$$$$$$$$$$
C  PROGRAMMER      ****** MAUNG SOE TINT ******
C                      $$$$$$$$$$$$$$$$$$$$$$$$$
C  PURPOSE : TO COMPUTE THE MAJOR KINEMATIC DESIGN VARIABLES
C  SUCH AS : LINK LENGTHS, OUTPUT ANGLES, PIVOT LOCATIONS,
C  SIZE AND OTHER RELATING MOTION CHARACTERISTICS.
C                      $$$$$$$$$$$$$$$$$$$$$$$$
C
C  PROGRAM DESCRIPTION.......
C  THIS PROGRAM COMPUTES THE BEST COMBINATION OF INPUT
C  LINK, COUPLER LINK FOR THE SPECIFIED REQUIRED OUTPUT
C  ANGLE AT BEST POSSIBLE TRANSMISSION ANGLE.
C  THE PROGRAM ALSO COMPUTES THE MINIMUM POSSIBLE
C  CAM SIZE FOR THE SPECIFIED FOLLOWER MOTION.
C                      $$$$$$$$$$$$$$$$$$$$$$$$
      REAL TH1,TH2,PH1,Q,P,K,X,FIRST,PX1,PX2,PY1,PY2,QX1,
     *QX2,L1,L2,M1,M2,N1,N2,A,B,C,D,E,F,WW,ANG,Z1,Z2,PH2,YDDP,
     *PH3,PH11,PH22,SS1,SS2,SI11,SI22,MU11,MU22,MU,MUU,S,T,TT,
     *MMU,TOL,REQANG,H,THETA,BETA,AA,BB,CC,DD,EE,FF,GG,KK,
     *LIMANG,LRDIAN,DIFF,R,THINT,THINT1,THINT2,PHINT,PHINT1,
     *PHINT2,DWD,WDW,RDR,SDS,ADA,BDB,CDC,FIDWEL,RR
      REAL QQ,AAA,BBB,Y,YD,YDMAX,YDMIN,YDD,YDDD,MAXL,NUMER,
     *DENOM,THEDEG,DNS,DN,YP,YDP,RAB,RAFOL,TANALP,ALPHAR,
     *EEOPT,RABMIN,PH3BIG,MALKIN,MUSTEL,GUILEN,NEGMAX,ALPHDG,
     *OVEHAN,THEDA,DNSP,U,V,XX,XXX,YY,DDEN,RAPITH,DNN,DOO,J,
     *RAPRIM,RAPRMI,OMEGA,LJ,LJDG,ALPHAM,RBFLAT,RBASE,RBASF,
     *YCOR,XCOR,THETAA,ARRAY(125,5)
      INTEGER I,N,ROW,COL
C
C
C       ********** DESCRIPTION OF VARIABLES **********
C
C
C     UNITS ARE LENGTH IN CENTIMETER AND ANGLE IN RADIAN
C     UNLESS OTHERWISE STATED.
C     TH1    .. INITIAL INPUT ANGLE.
C     THINT  .. INTERMEDIATE INPUT ANGLE.
C     PHINT  .. INTERMEDIATE OUTPUT ANGLE.
C     TH2    .. FINAL INPUT ANGLE.
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C     PH1      .. INITIAL OUTPUT ANGLE.
C     PH2      .. FINAL OUTPUT ANGLE.
C     Q        .. LENGTH OF DRIVEN LINK.
C     P        .. LENGTH OF INPUT LINK.
C     K        .. HORIZONTAL COMPONENT OF BASE LINK.
C     R        .. VERTICAL COMPONENTS OF CENTRE OF ROTATION
C                 OF INPUT LINK AND OUTPUT LINK.
C     X,FIRST.. CONTROL VARIABLES IN A LOOP.
C     PX1      .. HORIZONTAL COMPONENT OF THE INPUT LINK AT
C                 INITIAL POSITION.
C     PX2      .. HORIZONTAL COMPONENT OF THE INPUT LINK AT
C                 FINAL POSITION.
C     PY1      .. VERTICAL COMPONENT OF THE INPUT LINK AT
C                 INITIAL POSITION.
C     PY2      .. VERTICAL COMPONENT OF THE OUTPUT LINK AT
C                 FINAL POSITION.
C     QX1      .. HORIZONTAL COMPONENT OF THE OUTPUT LINK AT
C                 INITIAL POSITION.
C     QX2      .. HORIZONTAL COMPONENT OF THE OUTPUT LINK AT
C                 FINAL POSITION.
C     L1=L2    .. LENGTH OF COUPLER AT INITIAL AND FINAL
C                 POSITION.
C     M1       .. HORIZONTAL COMPONENT OF COUPLER LINK AT
C                 INITIAL POSITION.
C     M2       .. HORIZONTAL COMPONENT OF COUPLER LINK AT
C                 FINAL POSITION.
C     N1       .. VERTICAL COMPONENT OF COUPLER LINK AT
C                 INITIAL POSITION.
C     N2       .. VERTICAL COMPONENT OF COUPLER LINK AT
C                 FINAL POSITION.
C     AA       .. NUMERICAL VALUE OF TWO PI RADIANS.
C     BB       .. RECIPROCAL VALUE OF TWO PI.
C     A,B,C,D,E,F,Z1,Z2,ANG,AAA,BBB,RR,QQ,WW ARE ASSIGNED
C                 VARIABLES AS DESCRIBED IN THE PROGRAM.
C     I,N      .. LOOP COUNTER.
C     PH3      .. NET OUTPUT ANGLE IN DEGREES.
C     PH11     .. INITIAL OUTPUT ANGLE IN DEGREES.
C     PH22     .. FINAL OUTPUT ANGLE IN DEGREES.
C     SS1      .. TANGENT OF ANGLE OF COUPLER LINK AT INITIAL
C                 POSITION.
C     SS2      .. TANGENT OF ANGLE OF COUPLER LINK AT FINAL
C                 POSITION.
C     SI11     .. ANGLE MADE BY COUPLER WITH HORIZONTAL AT
C                 INITIAL POSITION.
C     SI22     .. ANGLE BETWEEN COUPLER AND HORIZONTAL AT
C                 FINAL POSITION.
C     MU11     .. ANGLE BETWEEN COUPLER LINK AND DRIVEN LINK
C                 AT INITIAL POSITION.
C     MU22     .. ANGLE BETWEEN COUPLER LINK AND DRIVEN LINK
C                 AT FINAL POSITION.
C     MU       .. INITIAL TRANSMISSION ANGLE.
C     MUU      .. FINAL TRANSMISSION ANGLE IN DEGREES.
C     S        .. DISPLACEMENT OF THE FOLLOWER WITH FUNCTION
C                 OF CAM ROTATION.
C     T        .. ANGULAR MOVEMENT OF CAM TO PITCH POINT (RAD)
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C     TT      .. ANGULAR MOVEMENT OF CAM TO PITCH POINT (DEG)
C     MMU     .. DIFFERENCE IN INITIAL AND FINAL
C                TRANSMISSION ANGLE IN DEGREES.
C     TOL     .. TOLERANCE IN DEGREE.
C     REQUANG . MINIMUM AMOUNT OF ANGLE IN DEGREE REQUIRED
C                BY THE CAM MECHANISM FOR THE SPECIFIED
C                FOLLOWER MOVEMENT.
C     H       .. MAXIMUM RISE OF THE FOLLOWER.
C     THETA   .. CAM ANGLE THROUGH WHICH THE CAM TURN FOR A
C                PARTICULAR TIME.
C     BETA    .. TOTAL CAM ANGLE REQUIRED FOR THE ENTIRE
C                RISE IN RADIAN.
C     CAMANG  .. CAM ANGLE THROUGH WHICH THE CAM TURN FOR A
C                COMPLETE RISE IN DEGREE.
C     EE      .. OFFSET DSTANCE.
C     MALKIN  .. MAXIMUM ALLOWABLE WORKING PRESSURE ANGLE IN
C                THE GUIDES.
C     Y       .. DISPLACEMENT OF THE FOLLOWER.
C     YD      .. CONVERTED VELOCITY OF THE FOLLOWER.
C     YDMAX   .. MAXIMUM CONVERTED VELOCITY OF THE FOLLOWER.
C     YDMIN   .. MINIMUM CONVERTED VELOCITY OF THE FOLLOWER.
C     YDD     .. CONVERTED ACCLERATION OF THE FOLLOWER.
C     YDDD    .. JERK OF THE FOLLOWER.
C     J       .. NUMERICAL VALUE OF TANGENT (30) DEGREES.
C     THEDEG  .. ANGLE SWEPT BY CAM TO PITCH POINT IN DEGREE
C     DNN     .. VERTICAL DISTANCE FROM CENTRE OF CAM TO
C                PITCH POINT.
C     DN      .. VERTICAL DISTANCE FROM CENTRE OF CAM TO
C                PRIME CIRCLE.
C     YP      .. DISPLACEMENT OF THE FOLLOWER TO PITCH
C                POINT.
C     YDP     .. CONVERTED VELOCITY OF THE FOLLOWER AT
C                PITCH POINT.
C     RAPITH  .. PITCH CIRCLE RADIUS OF CAM.
C     RAB     .. BASE CIRCLE RADIUS OF THE CAM.
C     RAFOL   .. RADIOUS OF THE FOLLOWER.
C     LIMANG  .. LIMITING PRESSURE ANGLE IN DEGREE.
C     TANALP  .. TANGENT VALUE OF PRESSURE ANGLE.
C     ALPHAR  .. PRESSURE ANGLE.
C     ALPHAM  .. MAXIMUM PRESSURE ANGLE IN RADIAN.
C     ALPHDG  .. MAXIMUM PRESSURE ANGLE IN DEGREE.
C     EEOPT   .. OPTIMUM OFFSET DISTANCE.
C     RBASF   .. BASE CIRCLE RADIUS OF CAM FOR FLAT-FACE
C                FOLLOWER.
C     RABMIN  .. MINIMUM POSSIBLE BASE RADIUS OF CAM.
C     RAPRIM  .. PRIME CIRCLE RADIUS OF CAM.
C     RAPRMI  .. MINIMUM POSSIBLE PRIME CIRCLE RADIUS.
C     PH3BIG  .. MAXIMUN OUTPUT ANGLE AVAILABLE FROM THE
C                FOUR BAR LINKAGE.
C     MUSTEL  .. STATIC FRICTION BETWEEN STEEL AND STEEL.
C     GUILEN  .. GUIDE LENGTH OF THE TRIP MECHANISM IN CM.
C     OVEHAN  .. OVERHANG DISTANCE OF TRIP MECHANISM.
C     THEDA   .. ANGLE MADE BY THE CAM TO THE POINT WHERE
C     OMEGA   .. ANGULAR VELOCITY OF CAM(CLOCK WISE)
C     YDV     .. VELOCITY OF THE FOLLOWER.
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C     RHOK    .. RADIUS OF CURVATURE OF CAM.
C     ROKHMI .. MINIMUM RADIUS OF CURVATURE OF CAM.
C     NEGMAX .. MAXIMUM NEGATIVE ACCLERATION OF FOLLOWER.
C     DD      .. VERTICAL DISTANCE FROM THE CENTRE OF CAM
C                TO THE CENTRE OF FOLLOWER AT ITS MINIMUM
C                LOWEST POSITION.
C     XX      .. X-COORDINATE OF CAM FROFIE.
C     YY      .. Y-COORDINATE OF CAM FROFILE.
C
C
C         ******** DESCRIPTION OF INPUT VARIABLES ********
      AA=6.2832
      BB=0.1592
      CC=39.4784
      LIMANG=30
      KK=2.8877
      H=1.4
      THETA=0.0
      TH1=0.38397
      THINT=0.6632
      TH2=1.2915
      PH1=0.5233
      Q=3.5
      P=3.4
      K=50.0
      R=12.5
      X=1
      FIRST=0.0
      PX1=COS(TH1)
      PX2=COS(TH2)
      PY1=SIN(TH1)
      PY2=SIN(TH2)
      QX1=COS(PH1)
      QY1=SIN(PH1)
      L1=60.0
      J=ATAN(LIMANG)
      CAMANG=100
      LRDIAN=30*0.017453
      BETA=CAMANG*3.1416/180
      PH3BIG=10E-02
      PRINT 50
      WHILE (P.LE.13.0 .AND. (L1+Q) .GE.(K-P*COS(TH2))) DO
        P=P+0.1
         M1=K-P*COS(TH1)+Q*COS(PH1)
         N1=R-P*SIN(TH1)+Q*SIN(PH1)
         L1=SQRT(M1**2+N1**2)
         L2=L1
         EXECUTE TANPHI
         IF (X .EQ. 0.0 ) THEN DO
            GO TO 555
         END IF
         EXECUTE OUTANG
         EXECUTE TRANGL
         PRINT 100,P,L1,PH22,MU,MUU
  555 CONTINUE
```

```
      END WHILE
      P=P-0.1
         PRINT 200,P,L1,PH3BIG
C
C     ******************************************************
C     * TO FIND THE INITIAL DWELL ANGLE IN DEGREE          *
C     ******************************************************
C
      ADA=2*R*Q-2*Q*P*SIN(THINT)
      BDB=2*K*Q-2*P*Q*COS(THINT)
      CDC=2*K*P*COS(THINT)+2*R*P*SIN(THINT)-2*R*P*SIN(TH1)-
     *2*P*SIN(TH1)*Q*SIN(PH1)+2*K*Q*COS(PH1)-
     *2*K*P*COS(TH1)-2*Q*COS(PH1)*P*COS(TH1)+2*R*Q*SIN(PH1)
      RDR=ADA**2-CDC**2
      SDS=BDB**2-CDC**2
      WDW=(2*ADA*BDB)**2-4*RDR*SDS
      DWD=SQRT(WDW)
      PHINT1=ATAN(((-ADA*BDB)+DWD)/(2*(ADA**2-CDC**2)))
      PHINT2=ATAN(((-ADA*BDB)-DWD)/(2*(ADA**2-CDC**2)))
      IF (PHINT1 .LT. 0.0) THEN DO
         PHINT=PHINT2*180/3.1416-PH11
         ELSE DO
         PHINT=PHINT1*180/3.1416-PH11
      END IF
      FIDWEL=PH3BIG-PHINT-CAMANG
      PRINT 220,PHINT,CAMANG,FIDWEL
  220 FORMAT('0',///,T5,' INITIAL DWELL DEGREE OF CAM IS',
     *2X,F12.9,2X,'EFFECTIVE CAM ANGLE IS',2X,F12.8,2X,
     *'FINAL DWELL ANGLE DEGREES IS',2X,F12.8///)
      EXECUTE ALPRES
      STOP
   50 FORMAT('1',T19,'--------------------------------------',
     *  '-----------------------------------------------'/
     *'-',T21,'DRIVER',15X,'COUPLER',11X,'OUTPUT',11X,
     *      'INITIAL TRANS',9X,'FINAL TRANS'/
     *      '0',T21,'LINK CM',14X,'LINK CM',11X,'ANGLE',12X,
     *      'MISSION ANGLE',9X,'MISSION ANGLE'/
     *      '0',T19,'--------------------------------------',
     *  '-----------------------------------------------'/)
  100 FORMAT('0',T20,F13.10,7X,F12.9,3X,F15.9,9X,F8.5,13X,F8.5)
  150 FORMAT('0','GOOD COMBINATION',2X,F13.10,7X,F12.9,3X,
     *      F15.9,9X,F8.5,13X,F8.5)
  200 FORMAT('0',///,T37,'*** THIS COMBINATION IS BEST FOR THE',
     *      1X,'REQUIRED MOTION ***'//
     *      '0',T20,'DRIVER LINK IS',F9.5,2X,'CM',3X,'COUPLER',
     *      1X,'IS',1X,F9.6,2X,'CM',3X,'OUTPUT ANGLE IS',3X,F9.5/
     *      ,T20,'############################################',
     *      '############################################')
  250 FORMAT('1',T17,'------------------------------------------',
     *      '-------------------------------------------'//
     *      T17,'DISPLACEMENT',6X,'VELOCITY',9X,'MAX VEL'
     *      ,10X,'MIN VEL',8X,'ACCLERATION',6X,'JERK'//
     *      T17,'------------------------------------------',
     *      '-------------------------------------------'//)
  300 FORMAT('0',T40,'*** THE TRIP MECHANISM TENTATIVELY',
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    *       2X,'DESIGNED ***',//
    *       ,T45,'* GIVES MAXIMUM ALLOWABLE PRESSURE ANGLE *'//
    *       ,T45,'* WITHOUT JAMING AS',3X,F8.5,3X,'DEGREE *')
350 FORMAT('0',T18,F8.4,8X,F8.4,6X,F11.4,5X,F11.4,7X,F8.4,
    *       10X,F8.4//)
400 FORMAT('0',///,T50,'*** FOR FLAT-FACE TYPE ***'//
    *       T30,'*** MINIMUM CAM SIZE OF BASE RADIUS'
    *       ,F8.4,1X,'cm IS FREE FROM CUSPS ***'///////
    *       '0',T30,'-----------------------------------',
    *               '-----------------------------------'//
    *       T52,'*** ITS PROFILE IS AS ***'//
    *       '0',T36,'THETA IS',15X,'X COORDINATE IS',8X,
    *       'Y COORDINATE IS'/
    *       '0',T30,'----------------------------------------',
    *               '--------------------------------'///)
450 FORMAT('0',T32,F12.7,14X,F12.7,11X,F12.7)
500 FORMAT(//T29,'ANGLE REQUIRED TO REACH PITCH POINT'
    *       ,1X,'IS',2X,F12.9,2X,'DEGREE'/
    *       ,T29,'====================================',
    *       '================')
550 FORMAT('0',T10,'*** THIS PITCH POINT CONDITION AT',2X,
    *       F12.9,2X,'DEGREE EXCEEDS MAXIMUM CONTROLLED',2X,
    *       'PRESSURE ANGLE ***'//)
600 FORMAT('0',///,T14,'*** THEREFORE NEGATIVE MAGNITUDE OF',
    *       'MAXIMUM PRESSURE ANGLE IN RADIAN',1X,F10.6,1X,
    *       'WAS SET UP ***'//
    *       T54,'+++ THIS TIME +++'//)
650 FORMAT('0',T8,'FOR THE QUIET OPERATION MINIMUM POSSIBLE',
    *       2X,'BASE RADIUS',2X,F8.4,3X,'WAS CHOSEN AT OFFSET'
    *       ,2X,F8.6/
    *       ,T8,'=========================================',
    *       '============================================',
    *       '=='//)
700 FORMAT('0',T20,'WHEN ANGLE IS',7X,'FOL RISE',13X,'FOL',
    *       'VEL',12X,'PRESSURE ANGLE'/
    *       T20,'----------------------------------------------',
    *       '----------------------------'//)
750 FORMAT('0',T23,I4,11X,F10.6,11X,F9.5,11X,F13.9///)
800 FORMAT('0',T12,'*** MINIMUM RADIUS OF CURVATURE',1X,F8.5
    *       ,2X,'CM OCCURS AT THETA IS',2X,F12.8,1X,'DEGREE **'/)
850 FORMAT('0',T25,'SPECIFIED ROLLER FOLLWER',1X,F5.3,1X,
    *       'IS TOO BIG'//
    *       ,T25,'** REDESIGN THE ROLLER AND CAM SIZE **'//)
900 FORMAT('0',T25,'** MINIMUM RADIUS OF CURVATURE IS',
    *       'GREATER',1X,'THAN ROLLER RADIUS **'//
    *       '0',T42,'*** NO UNDER CUTTING OCCURS ***'//
    *       ,T20,'** CAM SIZE WAS',1X,F17.12,2X,'CM ROLLER'
    *       'FOLLOWER',1X,'RADIUS',2X,F5.3,2X,'CM **'//
    *       ,T32,'*** OFFSET DISTANCE',2X,F11.8,2X,'CM WAS'
    *       ,1X,'APPROVED ***'////
    *       ,T23,'------------------------------------',
    *       '-------------------')
950 FORMAT('0',T24,'THETA IS',11X,'X COORDINATE IS',13X,
    *       'Y COORDINATE IS'/
    *       T23,'----------------------------------------',
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     *          '------------------'/)
 1000 FORMAT('0',T25,I3,14X,F12.7,16X,F12.7)
C
C   *********************************************************
C   *REMOTE BLOCK TANPHI CALCULATES THE TANGENT VALUE OF    *
C   *OUTPUT ANGLE AT FINAL POSITION.                        *
C   *********************************************************
C
      REMOTE BLOCK TANPHI
         A=K**2+Q**2+R**2+P**2-2*K*PX2*P-2*R*PY2*P
         B=2*K*Q-2*PX2*Q*P
         C=2*R*Q-2*PY2*Q*P
         D=C**2-(L2**2-A)**2
         E=2*B*C
         F=B**2-(L2**2-A)**2
         WW=(E**2-4*D*F)/10**6
         IF (WW .GE. 0.0) THEN DO
            W=SQRT(WW)*10**3
            Z1=(-E-W)/(2*D)
            Z2=(-E+W)/(2*D)
         ELSE DO
            X=0.0
         END IF
      END BLOCK
C
C   *********************************************************
C   * REMOTE BLOCK OUTANG CALCULATE THE VALUE OF OUTPUT     *
C   * ANGLE IN DEGREES.                                     *
C   *********************************************************
C
      REMOTE BLOCK OUTANG
      ANG=Z1
      I=1
      WHILE (I .LE. 2) DO
         IF (ANG .GE. 0.0) THEN DO
            PH22=ATAN(ANG)*180/3.14
         ELSE DO
            PH22=180-ATAN(ABS(ANG))*180/3.14
         END IF
      IF (PH22 .GT. FIRST) THEN DO
         FIRST=PH22
      ELSE DO
         ANG=Z2
      END IF
      I=I+1
      END WHILE
      END BLOCK
C
C   *********************************************************
C   * REMOTE BLOCK TRANGL DETERMINES TRANSMISSION ANGLES AT *
C   * INITIAL AND FINAL POSITION.                           *
C   *********************************************************
C
      REMOTE BLOCK TRANGL
         PH2=PH22*3.1416/180
```

```
      M1=K-P*COS(TH1)+Q*COS(PH1)
      N1=R-P*SIN(TH1)+Q*SIN(PH1)
      M2=K-P*COS(TH2)+Q*COS(PH2)
      N2=R-P*SIN(TH2)+Q*SIN(PH2)
      SS1=N1/M1
      SI11=ATAN(SS1)*180/3.1416
      SS2=N2/M2
      SI22=ATAN(SS2)*180/3.1416
      PH11=PH1*180/3.1416
      MU11=(PH11-SI11)
      MU22=(PH22-SI22)
      MU=MUU=0.0
      IF (MU11 .GT. 90) THEN DO
         MU=(180-MU11)
      ELSE DO
         MU=MU11
         IF (MU22 .GT. 90) THEN DO
            MUU=(180-MU22)
         ELSE DO
            MUU=MU22
         END IF
      END IF
C
C   * ********************************************************
C   * SELECTION OF THE BEST COMBINATION OF INPUT LINK COUPLER*
C   * LINK BASE LINK WHICH WILL GIVE THE REQUIRED OUTPUT     *
C   * ANGLE AT OPTIMUM TRANSMISSION ANGLE POSSILE            *
C   **********************************************************
C
    MMU=MU-MUU
    TOL=2.0
    REQANG=120
    PH3=PH22-PH11
    IF ((PH3 .GT.REQANG) .AND. (ABS(MMU) .LE. TOL)) THEN DO
       PRINT 150,P,L1,PH22,MU,MUU
    END IF
       IF (PH3 .GT. PH3BIG) THEN DO
          PH3BIG=PH3
       END IF
    END BLOCK
C
C   **********************************************************
C   * TO DETERMINE ALLOWABLE PRESSURE ANGLE WITH SPECIFIED   *
C   * PARAMETERS SUCH AS OVERHANG LENGTH AND GUIDE LENGTH,   *
C   * ETC.                                                   *
C   **********************************************************
C
    REMOTE BLOCK ALPRES
       MUSTEL=0.15
       GUILEN=5.5
       OVEHAN=7.5
       MALKIN=ATAN(GUILEN/(MUSTEL*(2*OVEHAN+GUILEN)))*
    *   180/3.1416
       PRINT 300,MALKIN
C
```

```
C  ********************************************************
C  * FOR THE FOLLOWER MOTION CYCLOIDAL CHARACTERISTICS GIVE *
C  * BEST PERFORMANCE.THEREFORE APPLYING CYCLOIDAL MOTION TO*
C  * DETERMINE THE DISPLACEMENT, CONVERTED VELOCITY,MAXIMUM *
C  * AND MINIMUM CONVERTED VELOCITY AND JERK FOR THE        *
C  * SPECIFIED MECHANISM.                                   *
C  ********************************************************
C
      PRINT 250
         YDMAX=0.0
         YDMIN=RBASE=10E03
         ROW=1
         THETA=0.017453
      WHILE (THETA .LE. BETA) DO
         Y=H*(THETA/BETA-BB*SIN(AA*THETA/BETA))
         YD=H/BETA*(1-COS(AA*THETA/BETA))
         IF (YD .GT. YDMAX) THEN DO
            YDMAX=YD
         IF (YD .LT. YDMIN) THEN DO
               YDMIN=YD
         END IF
         END IF
         YDD=H*AA/BETA**2*SIN(AA*THETA/BETA)
         YDDD=CC*H/BETA**3*COS(AA*THETA/BETA)
         PRINT 350,Y,YD,YDMAX,YDMIN,YDD,YDDD
         RBFLAT=Y+YDD
         IF (RBFLAT .LT. RBASE) THEN DO
            RBASE=RBFLAT
         END IF
         RBASF= -(RBASE)+0.4491
         XCOR=(RBASF+Y)*COS(THETA)-YD*SIN(THETA)
         YCOR=(RBASF+Y)*SIN(THETA)+YD*COS(THETA)
         THETAA=THETA*180/3.1416
         ARRAY(ROW,1)=THETAA
         ARRAY(ROW,2)=XCOR
         ARRAY(ROW,3)=YCOR
         ROW=ROW+1
         THETA=THETA=0.017453
      END WHILE
         PRINT 400,RBASF
           ROW=1
            WHILE (ROW .LE. CAMANG) DO
               PRINT 450,ARRAY(ROW,1),ARRAY(ROW,2),
     *         ARRAY(ROW,3)
               ROW=ROW+1
           END WHILE
C
C  ********************************************************
C  * TO DETERMINE THE CAM ANGLE(THETAP) FROM THE BEGINNING  *
C  * TO THE PITCH POINT                                     *
C  ********************************************************
C
 666     MAXL=10E04
         T=0.0
         WHILE ((MAXL .GT. J) .AND. (T .LE. BETA)) DO
```

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          T=T+0.017453
          NUMER=(AA*H/BETA**2)*SIN(T*AA/BETA)
          DENOM=(H/BETA)*(1-COS(AA*T/BETA))
          MAXL=NUMER/DENOM
       END WHILE
       T=T-0.017453
       TT=T*180/3.1416
       PRINT 500,TT
       YP=H*(T/BETA-BB*SIN(AA*T/BETA))
       YDP=H/BETA*(1-COS(AA*T/BETA))
       YDDP=H*AA/BETA**2*SIN(AA*T/BETA)
       EE=YDMIN
    RAFOL=0.4
    RABMIN=10E02
777 WHILE (EE .LE. YDMAX) DO
    RAPRIM=(((YDP-EE)/J-YP)**2+EE**2)**0.5
    RAB=RAPRIM-RAFOL
    THETA=0.0
       WHILE (THETA .LE. BETA) DO
          THETA=THETA+0.017453
          Y=H*(THETA/BETA -BB*SIN(AA*THETA/BETA))
          YD=H/BETA*(1-COS(AA*THETA/BETA))
          DN=(RAPRIM**2-EE**2)**0.5
          TANALP=(YD-EE)/(Y+DN)
          IF (ABS(TANALP) .GT. ABS(J)) THEN DO
             EE=EE+0.1
             GO TO 777
          END IF
       END WHILE
       DIFF=0.001
       IF (ABS(RAPRIM-RBASF) .LE. DIFF) THEN DO
          EEOPT=EE
          RABMIN=RAB
          PRINT 650,RABMIN,EEOPT
          GO TO 888
       END IF
       EE=EE+0.1
       END WHILE
       J=-J
     PRINT 550,TT
     PRINT 600,J
       GO TO 666
C
C   *********************************************************
C   * CALCULATION OF PRESSURE ANGLE AT ONE DEGREE INTERVAL OF*
C   * CAM MOVEMENT FOR THE ENTIRE MOTION OF THE RISE WITH THE*
C   * CHOSEN OPTIMUM OFFSET AND BASE RADIOUS.               *
C   *********************************************************
C
  888 THETA=0.0
      PRINT 700
      RAPRMI=RABMIN+RAFOL
      DN=(RAPRMI**2-EEOPT**2)**0.5
      H=1.4
      N=0
```

```
      OMEGA=-5
      WHILE (THETA .LE. BETA) DO
         Y=H*(THETA/BETA-BB*SIN(AA*THETA/BETA))
      YD=H/BETA*(1-COS(AA*THETA/BETA))
      YDV=YD*OMEGA
      TANALP=(YD-EEOPT)/(Y+DN)
      ALPHAR=ATAN(TANALP)
      ALPHDG=ALPHAR*180/3.1416
      PRINT  750,N,Y,YDV,ALPHDG
         THETA=THETA+0.017453
         N=N+1
      END WHILE
C
C   **********************************************************
C   * DETERMINATION OF MINIMUM RADIUS OF CURVATURE OF THE    *
C   * TENTATIVELY DESIGNED CAM AND COMPARE WITH THE RAIOUS OF*
C   * THE ROLLER FOLLOWER. IF RADIOUS OF THE ROLLER FOLLOWER *
C   * IS GREATER THAN THE MINIMUM RADIUS OF CURVATURE THEN   *
C   * THE RADIOUS OF ROLLER FOLLOWER WAS DECREASED BY 0.1 CM *
C   * AND REDESIGN THE SIZE OF THE CAM UNTIL NO UNDER        *
C   * CUTTING OCCURS.                                        *
C   **********************************************************
C
      THETA=0.017453
      NEGMAX=10E06
      WHILE (THETA .LE. BETA) DO
         YDD=H*AA/BETA**2*SIN(AA*THETA/BETA)
         IF (YDD .LT. NEGMAX) THEN DO
            NEGMAX=YDD
            THEDA=THETA
         END IF
         THETA=THETA+0.017453
      END WHILE
      S=H*(THEDA/BETA-BB*SIN(AA*THEDA/BETA))
      YD=H/BETA*(1-COS(AA*THEDA/BETA))
      DDEN=(RAPRMI+S)**2+2*(YD)**2-(RAPRMI+S)*NEGMAX
      RHOKMI=(((RAPRMI+S)**2+(YD)**2)**1.5)/DDEN
      PRINT  800,THEDA,RHOKMI
      IF (RHOKMI .GE. RAFOL) THEN DO
         PRINT  900,RABMIN,RAFOL,EEOPT
      ELSE DO
         PRINT  850,RAFOL
         RAFOL=RAFOL-0.1
         GO TO 777
      END IF
C
C   **********************************************************
C   * SINCE DESIGN DATAS SATISFY REQUIREMENT CAM PROFILE IS  *
C   * DEVELOPED ROLLER FOLLOWER=RAFOL,OFFSET DISTANCE=EEOPT  *
C   * AND BASE CIRCLE RADIOUS OF THE CAM = RABMIN.           *
C   **********************************************************
C
      DD=((RABMIN+RAFOL)**2-EEOPT**2)**0.5
      THETA=0.0
      N=0
```

```
      PRINT 950
      WHILE (THETA .LE. BETA) DO
         S=H*(THETA/BETA-BB*SIN(AA*THETA/BETA))
         YD=H/BETA*(1-COS(AA*THETA/BETA))
         U=(DD+S)*SIN(THETA)-(EEOPT+YD)*COS(THETA)
         V=(DD+S)*COS(THETA)+(EEOPT+YD)*SIN(THETA)
         XXX=RAFOL/(1+(U/V)**2)**0.5
         XX=EEOPT*SIN(THETA)+(DD+S)*COS(THETA)-XXX
         YY=(XX*U+(DD+S)*YD)/V
         PRINT 1000,N,XX,YY
         THETA=THETA+0.017453
         N=N+1
      END WHILE
      END BLOCK
      END
$ENTRY
```

Appendix E

DETERMINATION OF MAXIMUM ALLOWABLE PRESSURE ANGLE

The following procedure is used to determine the maximum allowable pressure angle without follower jamming (refer to Figure 7).

P = Total load acting on the cam, N

F = Normal reaction of cam, N

N_1 and N_2 are normal reactions in the guide on the follower stem, N

a = pressure angle, degrees

A = overhang length, mm

B = guide length, mm

U = coefficient of friction between follower stem and guide, unitless

$$\Sigma Fy = 0, \qquad F(\cos a)-U(N_1)-U(N_2)-P = 0 \tag{E-1}$$

$$\Sigma Mj = 0, \qquad N_1(B)-U(N_1)(D/2)+U(N_2)(D/2)-F(A)\sin a = 0 \tag{E-2}$$

$$\Sigma Mk = 0, \qquad N_2(B)-U(N_1)(D/2)+U(N_2)(D/2)-F(A+B)\sin a = 0 \tag{E-3}$$

The difference between $U(N_1)(D/2)$ and $U(N_2)(D/2)$ is very small and therefore can be assumed zero. Then -

$$N_1 = [A(F)\sin a]/B \tag{E-4}$$

$$N_2 = [(A+B)(F)\sin \alpha]/B \qquad [E-5]$$

Equating (E-1) and (E-4) gives:

$$P = F(\cos \alpha)-U(F)(A/B)(\sin \alpha)-U(N_2) \qquad [E-6]$$

Equating (E-5) and (E-6) and solving for P and F gives:

$$P = F[\cos \alpha-\{U(2A+B)/B\}\sin \alpha] \qquad [E-7]$$

Similarly,

$$F = P/[\cos \alpha-(U/B)(2A+B)\sin \alpha] \qquad [E-8]$$

In equation (E-8) F will be infinite if the denominator is equal to zero. This means that the follower will jam in its guide. The denominator set equal to zero gives:

$$\cos \alpha-(U/B)(2A+B)\sin \alpha = 0 \qquad [E-9]$$

Solving for the maximum allowable pressure angle without jamming in the guide gives:

$$\alpha(\max) = \tan^{-1}[B/\{U(2A+B)\}] \qquad [E-10]$$

Substituting the values B = 55 mm, A = 75 mm and U = 0.15 yields:

$$\alpha\,(\max) = 60.7 \text{ degrees}$$

Appendix F

DETERMINATION OF THE CAM SIZE WITH ROLLER FOLLOWER

A sample calculation for the determination of the size with a roller follower is given below (refer to Figure 15):

$$R = d + f(\emptyset) \qquad [F-1]$$

$$d = (R0^2 - E^2)^{0.5} \qquad [F-2]$$

$$R = J \cos \alpha + K \cos \beta \qquad [F-3]$$

$$E = J \sin \alpha - K \sin \beta \qquad [F-4]$$

where,

R = the instantaneous distance between pitch curve and the centre of the cam, mm

$f(\emptyset)$ = the lift of the cam, mm from the lowest position to the point where the angular movement of the cam is considered.

R0 = prime radius of the cam, mm

E = offset distance, mm

J = instantaneous radius of curvature of pitch curve, mm

K = the distance between the centre of curvature of the pitch curve and the centre of the cam, mm

α = pressure angle, degrees

β = angle as shown in Figire 15, degrees

$\emptyset$ = angular movement of the cam, degrees

From [F-1] and [F-3],

$$d + f(\emptyset) = J \cos \alpha + K \cos \beta \qquad [F-5]$$

Differentiating [F-4] and [F-5] with respect to $\emptyset$

$$f'(\emptyset) = -J(\sin \alpha)d\alpha/d\emptyset - K(\sin \beta)d\beta/d\emptyset \qquad [F-6]$$

$$0 = J(\cos \alpha)d\alpha/d\emptyset - K(\cos \beta)d\beta/d\emptyset \qquad [F-7]$$

For an infinitesimal rotation of the cam,

$$d\beta/d\emptyset = -1 \qquad [F-8]$$

Again, rearranging [F-7]

$$d\alpha/d\emptyset = [K(\cos \beta)/\{(J)\cos \alpha)\}]d\beta/d\emptyset \qquad [F-9]$$

Substituting [F-8] and [F-9] in [F-6] and solving with [F-3] and [F-4] yields:

$$\tan \alpha = [f'(\emptyset) + E]/[d + f(\emptyset)] \qquad [F-10]$$

Equation [F-10] is a general equation for the pressure angle with cam rotation in a clockwise direction. It shows that if offsetting is on the wrong side, the pressure angle will increase instead of decreasing. To obtain the correct offsetting the cam should turn counterclockwise. Then E = -E and the pressure angle equation [F-10] will becomes:

$$\tan \alpha = [f'(\emptyset) - E]/[d + f(\emptyset)]$$

[F-11]

It follows that:

$$a = \tan^{-1}[f'(\emptyset) - E]/[d + f(\emptyset)]$$

[F-12]

Equation (F-11) represents a pressure angle equation with offset distance E. To determine the cam size for a specified maximum pressure angle, differentiate equation [F-11] with respect to time and equate to zero. The result is:

$$\sec^2 a \; da/dt = d(MM/NN)/dt$$

[F-13]

where,

$$MM = f'(\emptyset) - E$$

$$NN = d+f(\emptyset)$$

From [F-13], dividing by $\sec^2 a$ and completing the differential on the right hand side, gives:

$$da/dt = W(\cos^2 a)(AA/BB)$$

$$= 0$$

[F-14]

where,

$$AA = f''(\emptyset)\{f(\emptyset)+d\}-f'^2(\emptyset)-\{f'(\emptyset)\}E$$

$$BB = [d+f(\emptyset)]^2$$

W = angular velocity of cam, rad/s

Since, $(W)\cos^2 a/[d+f(\emptyset)]^2$ does not equal zero it follows that,

$$AA = 0$$

$$d+f(\emptyset) = [f'^2(\emptyset)-f'(\emptyset)(E)]/f''(\emptyset)$$

[F-15]

At the maximum pressure angle condition, for the radial cam, the offset distance, E is equal to zero and $d+f(\emptyset)$ becomes the radius of the pitch curve, Rp. It follows that:

$$Rp = f'^2(\emptyset)/f''(\emptyset)$$

[F-16]

$$\tan \alpha(max) = f'(\emptyset)/Rp$$

Or,

$$\tan \alpha(max) = f''(\emptyset)/f'(\emptyset)$$

[F-17]

In the offsetting condition $d+f(\emptyset)$ becomes the vertical distance from the centre of the cam to the pitch point, DNp. It follows that:

$$DNp = [f'^2(\emptyset)-f'(\emptyset)E]/[f''(\emptyset)]$$

[F-18]

$$\tan \alpha(max) = [f'(\emptyset)-E]f''(\emptyset)/[f'^2(\emptyset)-f'(\emptyset)E]$$

$$\tan \alpha(max) = [f''(\emptyset)]/[f'(\emptyset)]$$

[F-19]

For cycloidal motion,

$$f'(\emptyset) = H[1-\cos(2pi\ \emptyset p/\beta)]/\beta$$

[F-20]

$$f''(\emptyset) = 2H(pi)[\sin(2pi\ \emptyset p/\beta)]/\beta^2$$

[F-21]

where $\emptyset p$ = angle locating a pitch point, degrees

Substituting these expressions into equation (F-19) leads to an unmanageable transcendental equation. The problem was therefore solved using a high-speed computing machine. Since $a(max)$ has two values, $\emptyset p$ will have two values, respectively. Substituting these values in (F-11) and rearranging, yields

$$RO = [\{KK/(\tan a) - f(\emptyset)p\}^2 + E^2]^{0.5}$$

[F-22]

where,

$$KK = \{f'(\emptyset)p - E\}$$

For minimum possible cam size, the offset distance and the base diameter were determined as:

Base diameter of cam = 33.45 mm

Offset distance E = 10 mm

Checking for an undercutting condition-

To calculate the radius of curvature of the cam differentiate equation [F-11] with respect to $\emptyset$ and solve equations

[F-2],[F-6] and [F-11] to give:

$$J = FF/GG$$

[F-23]

where,

$$FF = [\{RO+f(\emptyset)\}^2 + f'^2(\emptyset)]^{1.5}$$

$$GG = \{RO+f(\emptyset)\}^2 + 2f'^2(\emptyset) - \{RO+f(\emptyset)\}\{f''(\emptyset)\}$$

Equation [F-23] is general for the radius of curvature of a synthesized cam. In equation (F-23) the minimum radius of curvature of the cam will occur where Ø gives the maximum negative acceleration. That is f''(Ø) = maximum negative value. From the nature of cycloidal motion the maximum negative acceleration will occur at 75 percent of maximum cam angle. Since the cam angle is 100 degrees, the maximum negative acceleration will occur when the cam turns 75 degrees from the end of the dwell position. Substituting this value into equation [F-23] yields:

Rho (min) = 18.39 mm

This value is greater than the roller follower radius of 4 mm. Therefore no undercutting or sharp corner will be present.

Appendix G

ANALYSIS FOR FOLLOWER SPRING DESIGN

In this design the only external load is the frictional force required to engage the tooth in the roller chain. The exact magnitude of the external load varies depending on whether or not the tooth hits a roller on the chain before engaging between the rollers. Mathematically, a force balance for the follower can be expressed as follows when the inertia force is about to lift the follower from the cam during deceleration of the cam:

$$F(sp) = F(in) - W(fol) - W(ex) - F(fri) \qquad [G-1]$$

where,

F(sp) = spring force, N

F(in) = inertia force due to maximum negative acceleration, N

W(fol) = force due to weight of the follower, N

W(ex) = external load, N

F(fri) = frictional resistance force developed in guides by stem, N

But, F(in) = W(fol)(a), and

F(fri) = U(Fn)sin α

where,

a = maximum negative acceleration, m/s^2

U = coefficient of friction between follower stem and guide, unitless

Fn = normal force exerted by the cam to the follower, N

a = maximum pressure angle produced by redesigned cam, degrees

$$Fn = P/DEN$$

where, $P = F(in) + H(neg)(Ss) + IL + W(fol) + W(ex)$

[G-2]

where,

H(neg) = rise of the follower from the lowest position to the point where maximum negative acceleration of the follower occurs, m

Ss = stiffness of the spring, N/m

IL = initially applied load to the spring (pre load), N

$$DEN = \cos a-(U/B)(\sin a)\{2A+B-U(D)\}$$

[G-3]

where,

B = guide length, m

A = overhang length, m

D = diameter of the follower stem, m

When assembling the compression spring on the follower it is common practice to initially compress the spring so that the roller follower will always contact the cam and thereby minimize backlash. Therefore, the spring stiffness Ss can be calculated as:

$$Ss = [F(sp) - IL]/H(neg)$$

[G-4]

The external load, W(ex) would exist only momentarily at the beginning of the rise of the follower. Its value would be zero at the 3/4 point of the cam angle in this mechanism. The spring stiffness can be expressed as:

$$Ss = [K-U(Q/DEN)\sin a-IL]/[H(neg)(1+U \sin a/DEN)]$$

where,

$$K = F(in) - W(fol)$$

$$Q = F(in) + W(fol) + IL$$

The appropriate sustitutions result in Ss = 540.4 N/m

POWER REQUIREMENT FOR THE TRIP MECHANISM

The tangential force acting horizontally on the pitch curve of the cam is given by:

$$F\sin\alpha = (P/DEN)\sin\alpha$$

$$= [\{Q+H(neg)(Ss)\}/DEN]\sin\alpha$$

$$= 11.98 \text{ N}$$

The maximum torque (N-m) required for the calculated load will be,

$$T = (F\sin\alpha)\{R0+H(neg)\}$$

where, R0 = prime circle radius of cam, m

Then, torque will be:

$$T = 0.4108 \text{ N-m}$$

Then maximum power required will be equal to (T)(W) where W is the angular velocity of the cam (rad/s).

where,

W = angular velocity of cam, rad/s

Substituting the appropriate values result in an estimate of the power requirement of 2 watts.