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A SIX - LINK GRIPPER FOR AN INDUSTRIAL ROBOT

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Abstract.

A compact and rugged gripper is introduced that has linear input. The gripper comprises two centric slider-crank mechanisms that are driven by a common crank. The sliders, to which the twin fingers are rigidly attached, run on the same axis. The gripper enables concentric grasping and force-intensification at the grip by the use of toggle action over a wide range of operation

Analytical studies leading to the final design are summarized. Evolution of the overall design is recapitulated. The final design is introduced, and the resulting gripper is presented.

1. Introduction

An industrial robot consists basically of a body or base, an arm, and a wrist. The gripper is attached to the wrist, and it enables the robot to pick up work pieces and hold, manipulate, transfer, place and release them accurately in a discrete position. Thus the gripper is an important interface between the robot and its environment without which the robot cannot function effectively. Custom-designed grippers can be quite costly. Surveys of grippers in current use are available [1-6].

The gripper for industrial robots is generally used to handle only one or a few objects of similar shape, size, and weight in a repetitive operation which requires minimum gripping dexterity, and is limited in its versatility. Where the robot is required to manipulate a tool rather than a work piece, the gripper is designed to grasp and handle the tool while at the same time supplying it with the required

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power to perform its function. This permits the robot to handle more than one tool during one cycle, often in conjunction with a quick-change device. In most applications, such as in spot and arc welding, spray painting, rotating spindles for metal cutting operations, heat torches, and water jet cutting, the tool is directly attached to the wrist of the robotic arm, and effectively becomes the gripper.

Grippers can be classified as a) mechanical finger type, b) vacuum and magnetic type, and c) universal grippers. The most common grippers today, in the order of use, are pneumatically actuated finger type, vacuum type, electrically actuated finger type, and hydraulically actuated finger type grippers [7]. Pneumatic systems are popular especially because they are easy to use, install, and service.

When used as a hand, the gripper is used to grasp by fingers, which generally are two non-jointed tongs. The fingers (jaws) are the appendages that make actual contact with the object. The grip in this case will be at a point [8]. Alternatively each of the two pivoted fingers may be made in the form of a parallel linkage, in which the coupler is formed in the shape of a grip head, resulting in a line contact. In grasping, the part is held (constrained) by the gripper by either shaping the fingers so as to enclose the part, or by friction. The latter case yields simpler finger geometry but requires higher grasping forces. Grippers with jointed fingers are relatively uncommon in industrial robotics.

Actuation of fingers can be by pneumatic or hydraulic actuators, by rope, or by rack-and-pinion arrangements, where the rack may be actuated by a solenoid. Wire-and-pulley mechanisms are used for master-slave type of manipulators [9]. Electric grippers generally use AC/DC motors or solenoids to activate the jaws; their use, however, is limited due to the weight penalty that they impose. Hydraulic grippers can exert large forces, and yet generally carry an excessive weight penalty also [7].

If the objective of the gripper is to pick up and move an object of known shape and size, it is possible to design a gripper with a single-degree of freedom that will meet specific requirements. The versatility of a given gripper can be improved if various designs of *finger(s)* can be incorporated to meet mission constraints. Certain applications may necessitate the changing of the gripper as a whole. The mechanism of grasping, the stability of the grip (grab), the safety and integrity of the gripped object, and planning and control of the grasp forces have been the subject of considerable theoretical and experimental investigations because of their importance in assembly operations.

Gvozdev [10] and Bicchi and Balkema [11] discussed the design of grippers that are equipped with means of intensifying the clamping force. Various design alternatives for grippers were advanced by Mannaa and associates [5, 12-14]. These grippers featured centric gripping and force intensification.

In what follows, we introduce a novel concentrically gripping gripper which allows force intensification.

2. The Gripper

The six-link gripper being presented here is a compact mechanism with linear input. Shown schematically in Fig. 1, it comprises two identical centric slider-crank mechanisms that are driven by a common crank. Crank A_oAC is pivoted at A_o. Sliders B and D, to which the twin fingers are rigidly attached, run on the same axis. The fingers may be straight-faced, as shown in Fig. 1, or they may be provided with V-notched faces, depending on the application. The shaded circle resembles the work piece. A linear actuator, not shown in Fig. 1, is normally attached to point A of the mechanism to drive the gripper.

A loop-closure equation may be written to study the behavior of the gripper. Let A_oB=x, AB=l_8 and A_oA=l_7. Considering a right-handed orthogonal coordinate system with the x-axis horizontal and positive to the right, we have the equation

$$\bar{i} + I_8 \left(\cos\theta_8 i + \sin\theta_8 j\right) = I_7 \left(\cos\theta_7 i + \sin\theta_7 j\right) \tag{1}$$

where the half jaw opening (distance) x and angle θ_8 are unknown for given values of crank angle θ_7 and link dimensions l_7 and l_8.For a crank length l_7 of unity, the following scalar equations may be written from vectorial Eqn 1:

$$A = I_8 \sin\theta_8 - \sin\theta_7 \tag{2}$$

and

$$B = x + I_8 \cos\theta_8 - \cos\theta_7 \tag{3}$$

Opting for an iterative solution for the unknowns x and θ_8, A and B in Eqns 2 and 3 represent errors for initial estimates for x and θ_8 Linearizing Eqns 2 and 3 by the use of a Taylor series expansion, it may be shown that corrections $\Delta\theta_8$ and Δx to θ_8 and x, respectively, are

$$\Delta\theta_8 = -\frac{A}{l_8 \cos\theta_8} \tag{4}$$

and

$$\Delta X = -B + l_8\left(\sin\theta_8\right)\Delta\theta_8 \tag{5}$$

Iterations may be terminated when the absolute values of the corrections become smaller than a predetermined small number. Otherwise initial estimates are revised by the use of the corrections, given by Eqns 4 and 5, and the next iteration is started by going back to Eqns 2 and 3.

Equation 1 was solved in the manner outlined above. Figure 2 illustrates the variation of non-dimensional jaw opening, $l = 2x / l_7$, with crank angle θ_7 and the ratio of lengths of the connecting rod and the crank arm, l_8 / l_7. It follows from Fig. 2 that jaw opening decreases with increasing crank angle until θ_7 reaches 180 degrees. This observation is true for all values of l_8 / l_7. The decrease in jaw opening is rapid at first, until a crank angle of about 95 degrees. The change in jaw opening becomes increasingly less with further increases in crank angle, until it becomes insignificant after about 130 degrees. It is interesting to note that increasing the l_8 / l_7 ratio causes a linear increase in the jaw opening at any crank angle.

The transmission angle [15] is a reliable measure of the "goodness" or smoothness of operation of a mechanism. For a slider-crank mechanism that is driven by its crank, as it is in the present case, the transmission angle is defined as the angle between the normal to the axis of the slider and the connecting rod. In general the transmission angle will be less than 90^o, the best possible value for a transmission angle being 90^o In terms of the present variables it may be defined as θ_8 minus 90^o.

Figure 3 shows the variation of the transmission angle in the mechanism of Fig. 1. In this figure, the transmission angle is mapped against the crank angle and the ratio of lengths l_8 / l_7. The smallest values of the transmission angle are observed to occur at a crank angle of 90^o as would be expected. Although this observation is very vivid for small values of l_8 / l_7, reduction in transmission angle is not so obvious for values of $l_8 / l_7 > 3$, when the transmission angles remain above 70^o for all crank angles of interest.

Practice has shown that a minimum transmission angle of 40^0 or greater will ensure the smooth operation of a slider-crank mechanism when inertia effects can be disregarded [15]. On the other hand, the operation of the gripper can be considered to be quasi-static. It would be safe, therefore, to opt for a minimum transmission angle of 40^0 Referring to Fig. 3, it may be concluded that a value for l_8 / l_7 that is equal to or greater than 1.22 will provide a transmission angle larger than 40^0 throughout the region of operation of the gripper. For design purposes, therefore, an l_8 / l_7 ratio of 1.35 may be assumed.

Drawing free body diagrams of the crank and connecting rods of the gripper (Fig. 4), and writing equations of equilibrium, it may be shown [16] that

$$M = -2\,x\,F_1 \tan\theta_8 \tag{10}$$

$$N = -F_1 \tan\theta_8 \tag{11}$$

where F_1 is the gripping (clamping) force to be exerted by the finger on the workpiece, N is the force that is exerted normal to the guide of the slider, and M is the torque that must be applied about A_o (Fig. 1). Figure 5 depicts the change in crank torque M with crank angle and ratio of lengths l_8/ l_7 when $F_1 = 100$ N. It may be observed from this figure that the torque requirement is fairly uniform with changes in the l_8/ l_7 ratio. This is true for all values of the ratio except for l_8/ l_7 values less than 1.5, when there is a moderate rise in the torque requirement at crank angles less than 100^0 followed by a reduction at larger crank angles. Figure 5 makes it clear, however, that the primary change in M occurs with changes in crank angle. Thus M is less than 250 units for crank angles less than 100^0 for $l_8/ l_7 > 1.5$. The torque requirement decreases nearly linearly with further increases in crank angle. It is clear that the torque actually becomes negative when the toggle position is crossed at 180^0.

The variation of normal force N acting on the guide of the slider, as computed by Eqn 11, is depicted in Fig. 6, as it varies with changes in crank angle and the ratio of l_8/ l_7. This figure makes it apparent that the normal force is essentially stagnant over almost all of the region of operation of the crank and for nearly all values of l_8 / l_7. The only exception to this observation is a narrow region straddling the 90^0 position of the crank, when, at one point, the normal force reaches a peak of nearly 350 newtons for $l_8/ l_7 < 1.5$.

3. The Gripper Assembly

Figure 7 shows two views of the assembly drawing of the six-link gripper. The gripper is actuated by a pneumatic actuator, denoted by 1 in the figure . The actuator is anchored on the frame (8) of the gripper by a trunnion (14), while the tip of its plunger (13) is pin-jointed to the extension of the common crank (2) of two slider-crank mechanisms. A specially contoured connecting rod (3) is pinned to one of the sliders, while the other connecting rod (4) is not contoured. One finger (6) is fixed to each slider. The common guide (11) of the two sliders is specially machined so as to prevent rotation while allowing sliding. The guide is supported at the common crank shaft (15) as well as at both its ends (12). The connecting rods are fashioned in pairs to ensure smoothness of operation.

The crank arms are specially contoured to avoid obstructions, and they are provided with bushes and bosses at several joints to allow rotation only about desired axes. Connections at joints are made by the use of C-clips (snap rings) for compactness. The U-shaped frame of the gripper is a square pipe. A keyed bush complete with a set screw is welded to the frame for attachment of the gripper to the output shaft of the gearhead of motor #1. Holes are machined at the tips of the U for accommodating the end bushes of the slider guide.

4. Concluding Remarks

A basic requirement related with the smoothness of operation of the gripper, expressed in terms of transmission angles, will be satisfied when a ratio l_8 / l_7 of 1.35 is adopted. It may be shown for this particular case that the minimum transmission angle will be about 42^0, which is quite satisfactory.

To test the feasibility limits of the gripper, let it be assumed that the same gripper will be utilized for work pieces ranging in diameter from 70 mm to 130 mm. It is to be pointed out that a range of operation of such magnitude is clearly outside the operational limits of most commercial grippers. Using a gripper of appropriate dimensions, it turns out that a work piece 100 mm in diameter will be gripped at a crank angle of 125^0, corresponding to a transmission angle of 53^0. The largest-diameter work piece (130 mm) will be gripped at a crank angle of about 107^0, which features an admissible transmission angle of 45^0.

If a 5-kg payload is to be gripped between two smooth parallel faces of flat jaws, the force F_1 (Fig. 4) that must be applied for a safe grip depends on the coefficient of friction between the gripper surface and the surface of the work piece. For a friction coefficient of 0.2 between two steel surfaces, it may be shown that F_1 must be 125N. Let it be assumed, for the safety of the grip, that a larger gripping force is provided, such that $F_1 = 300$N. It may be verified that the corresponding torque requirement for the gripper of Fig. 7 ranges from 54 N-m at a crank angle of 94^0, when a workpiece 130 mm in diameter is being gripped, to zero at 180^0 The corresponding normal force N (Fig. 4) is initially about 328N. The normal force decreases rapidly with increasing crank angle, until it vanishes at the toggle position of 180^0

In conclusion it may be stated that the construction of the gripper is rugged and its design is simple. The unique six-bar design of the gripper enables concentric gripping. The gripper enables force-intensification at the grip by the use of toggle action over a wide range of operation.

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ذراع قابض سداسي لرجل آلي صناعي

اكورت، الجاوي، بوغوص، الجنيدي والدهلاوي

ملخص

تم تطوير تصميم محكم ومتين لرجل آلي يعمل بمدخل احادي يتكون التصميم من مجموعة أعمدة وزلاجة مركزية يعملان عن طريق عامود مشترك. الزلاجة المثبت عليها الاصبعين التوأم تعمل على نفس المحور. يمكن هذا التصميم من المسك وتركيز القوة القابضة .

تم تلخيص الدراسات التحليلية التي ادت الى التصميم النهائي لهذا الرجل الآلي الصناعي.

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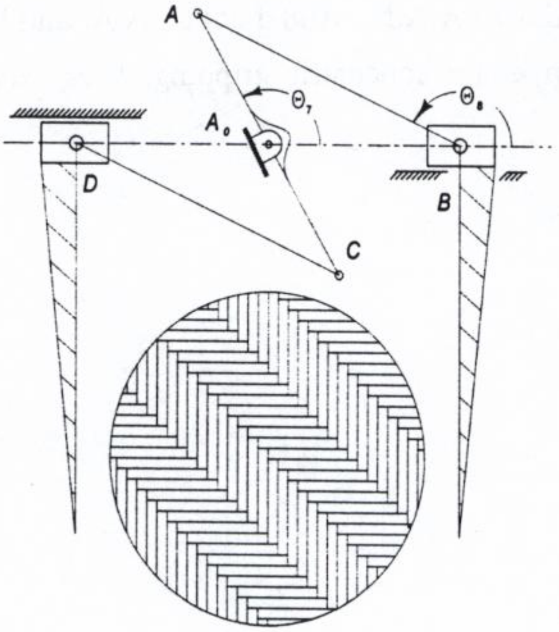

Fig. 1. A schematic view of the gripper.

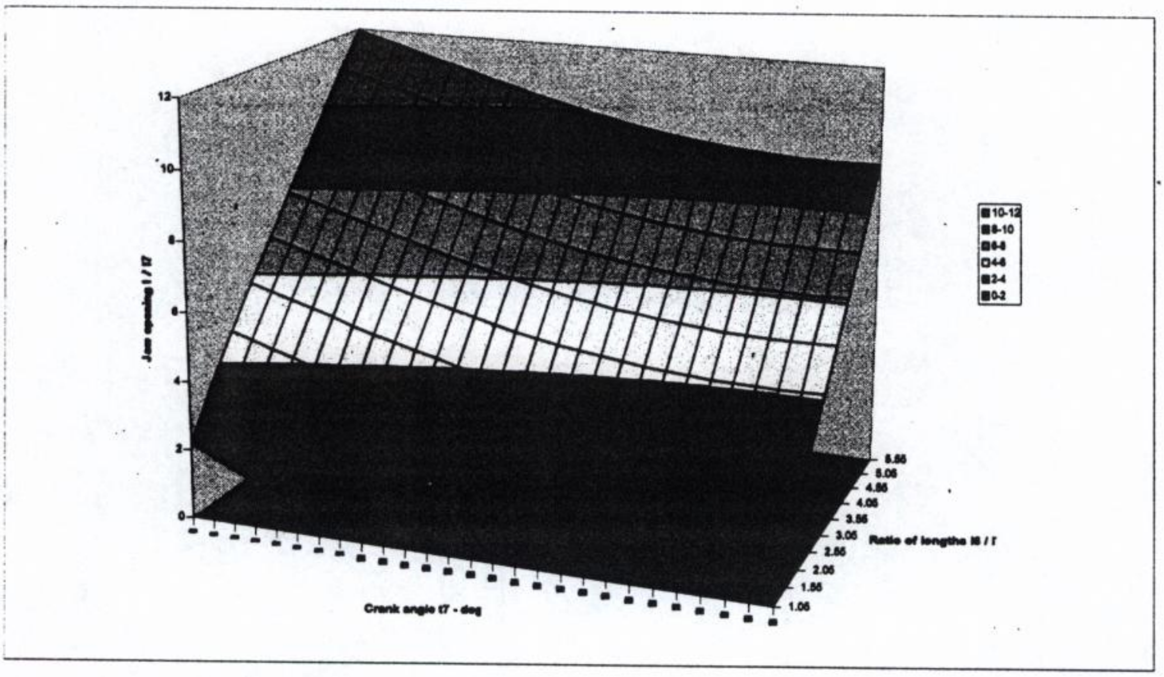

Fig. 2. Variation of jaw opening L with crank angle and ratio of length L_8/L_7.

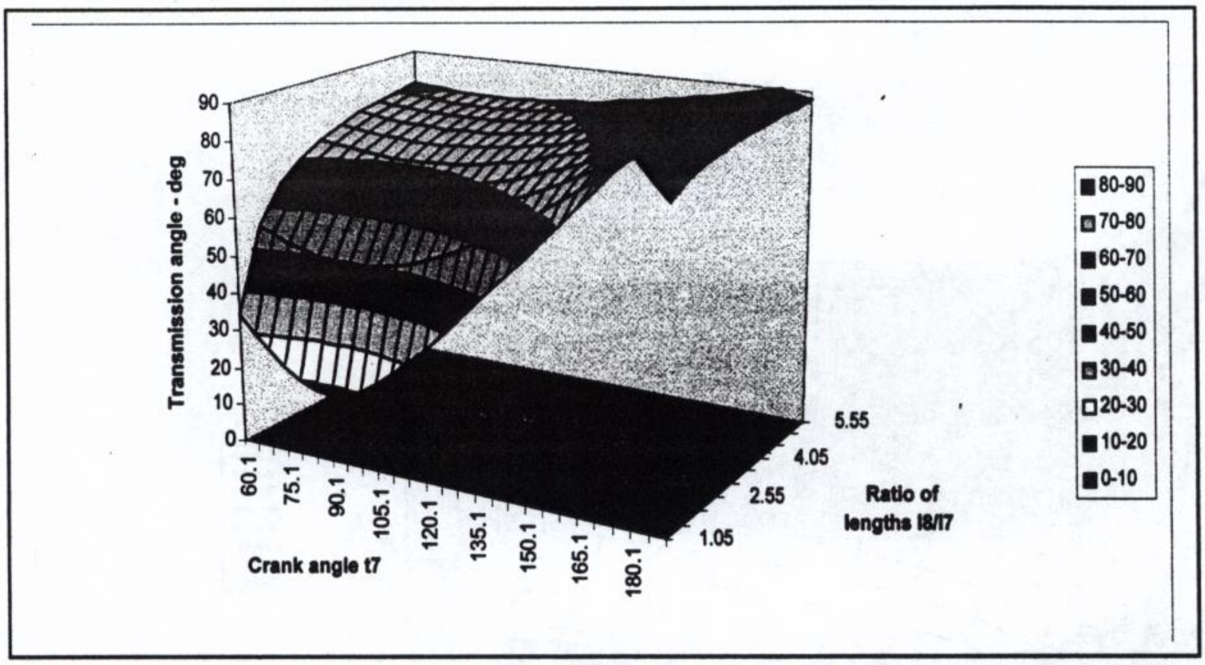

Fig. 3.1 : Change in transmission angle with crank angle and ratio of length L_8/L_7.

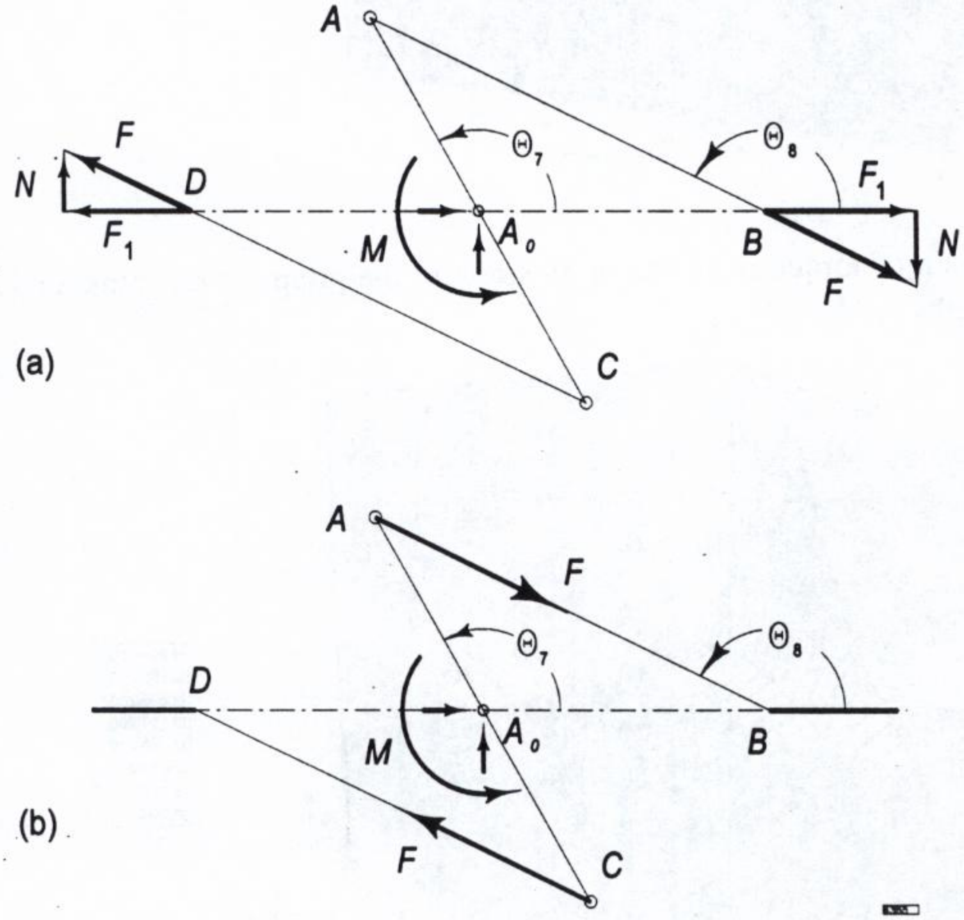

Fig. 4. Free body diagrams of the gripper.

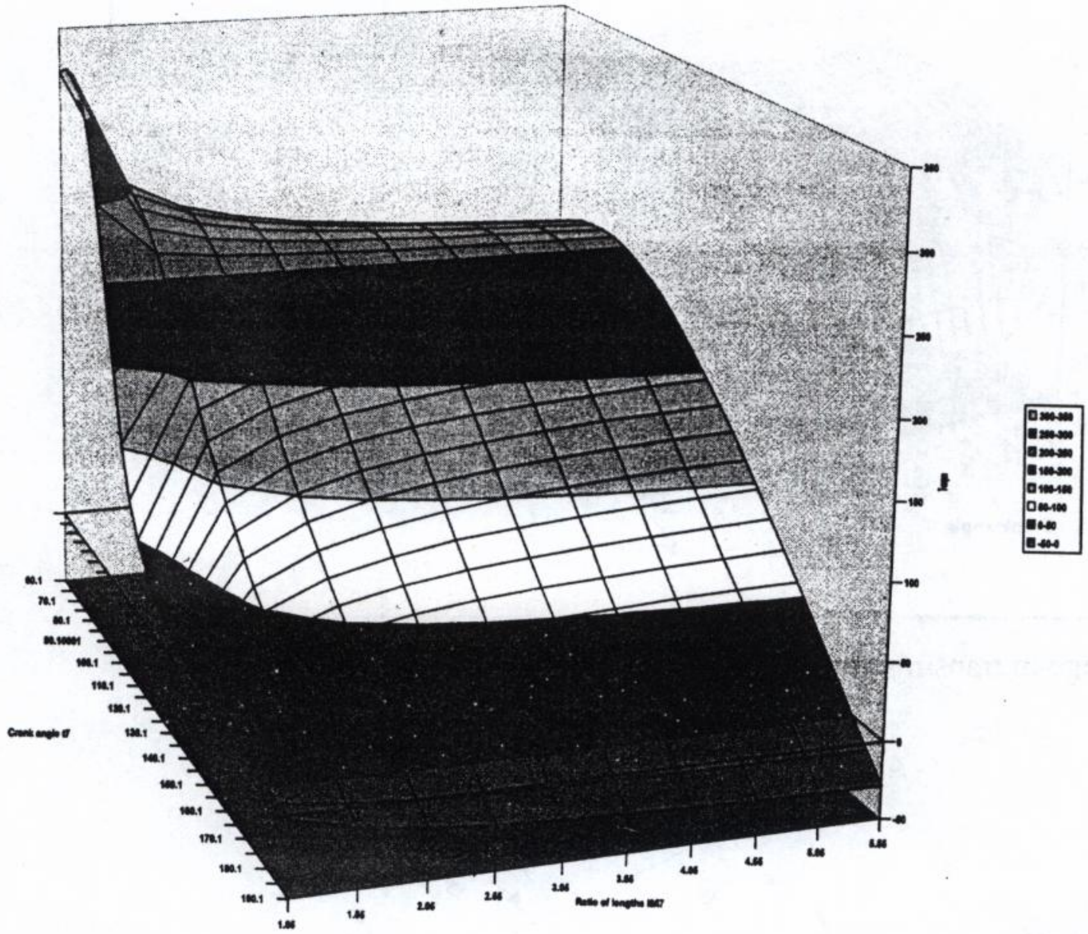

Fig. 5. 1 Variation of torque requirement on crank of the gripper with crank angle and ratio of lengths l_8 / l_7.

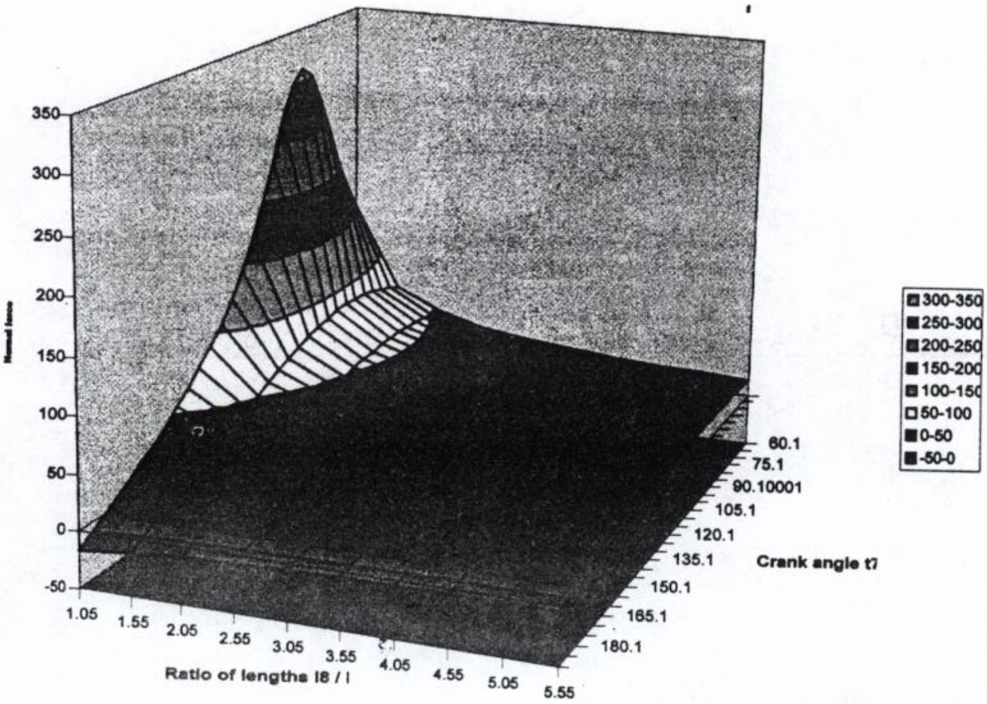

Fig. 6.1. Variation of normal force on slider with crank angle and ratio of lengths l_8 / l_7.

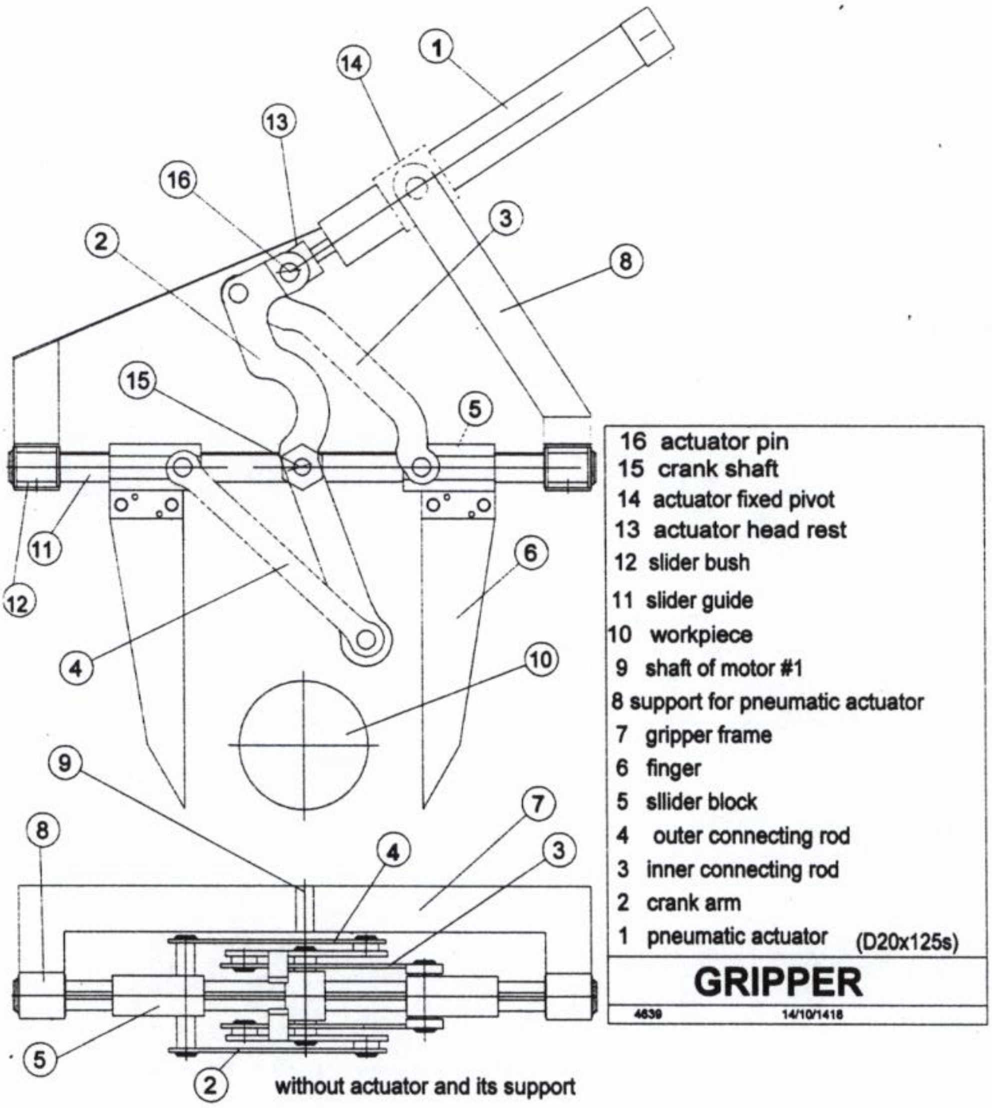

Fig. 7. The pneumatically actuated gripper.