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CAM-FOLLOWER MECHANISM DESIGN FOR NARROW LOOM BEAT UP MOTION

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ABSTRACT

A cam swing roller-follower mechanism is designed for the beat-up motion of a horizontal narrow loom. The system consists of a radial plate-cam driven by a camshaft keyed to the plate cam. A slay bar which act as the beater is attached to the radial swing roller-follower and assembled on the plate cam. A continuous contact between the roller follower and the plate-cam is maintained by a spring attached to the follower and fixed to a reference frame to prevent mechanism bounce. The conceptual design for the mechanism is based on the fundamental generalized synthesis procedure. The topology of the kinematics is developed by using the graph theory method of kinematic synthesis. The forces required to drive the plate-cam and follower system were modeled and the components such as the plate-cam, camshaft, the follower and the drive mechanism were synthesized for smooth operation of the mechanism. Analysis of the force requirement show that maximum impact required to beat the weft into the yarn during weaving is achieved at the maximum plate-cam displacement. The acceleration of the cam plate is controlled in the model at the start and end of motion as boundary conditions to specify some degree of stability for the system.

KEYWORDS: Mechanism, Follower, Cam-plate, Beat up, Narrow loom.

1. INTRODUCTION

Beat up is one of the principal operations in the weaving process in a narrow horizontal loom. The other motions include the shedding motion, let off motion, the pickup motion and take up motion. The beat up motion beats the weft into the warp yarn. The weaving sequence and rhythm of these motions are discussed in Raji, 2000. Several attempts have been made in the area of mechanizing and automating the operation of the horizontal weaving loom. Raji, 2000 developed a mathematical model which described the pattern of motion of the beat up action of the weaving loom. The type of motion found to be close to Simple Harmonic Motion (S. H. M) is developed from standard parametric polynomials by controlling the boundary acceleration of the primitive beater. The profile for a plate cam which exhibits the motion was developed in Raji and Adegbuyi, 2003. Several mechanisms have been described for transmitting oscillating motion from one plane to another at right angles to it. A means for providing the interrupting control of this oscillating movement is discussed in Staescu and Samer, 2008. The equation for the cutter point coordinates for a plate cam was derived to obtain the cam profile for the specified beater motion. Recep et. al., 2005 developed mechanism model for rotary dobby, crank and cam shedding motions where equations governing heald frame motion for shedding are derived, heald frame motion curves are obtained and a rotary dobby mechanism by computation was proposed and preferred to a crank and cam shedding mechanism due to the intermittent nature of the rotary dobby shaft's motion.

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This paper discussed the design analysis of a radial cam roller swinging follower mechanism which could suitably replace the primitive beat up mechanism of the horizontal weaving loom.

Principle of Operation

During full rotation of the camshaft, the plate cam rotates to dwell for half of the rotation, rises for a quarter and fall for the next quarter to complete the motion. The period of dwell of the cam is to allow for the picking ad shedding motion of the loom. The pattern of motion is traced by the follower which is keyed to a follower shaft carrying the slay bar. The resulting motion of the follower is transmitted to the beater via the slay bar.

2. DESIGN AND MODELING

Figure 1 shows a schematic view of the system. The conceptual design of the mechanism is based on the generalized synthesis procedure as discussed in Kota and Chiou, 1992. The ability to enumerate all possible kinematic topology using graph theory as discussed in Kannapan and Marshek, 1990, Kramer and Premkumar, 1988 and Thompson et al., 1985 lends itself to the development of the mechanism. The cam–follower system consists of the radial plate cam driven by a cam shaft keyed to the cam plate. A radial swinging follower is assembled on the plate cam and carries a slay bar on which the beater reed is fixed. The continuous contact of the follower and the plate cam profile surface is ensured by a return spring attached to the follower and fixed to the system frame. The mechanism is designed to replace the beater swinging arrangement shown in Figure 2.

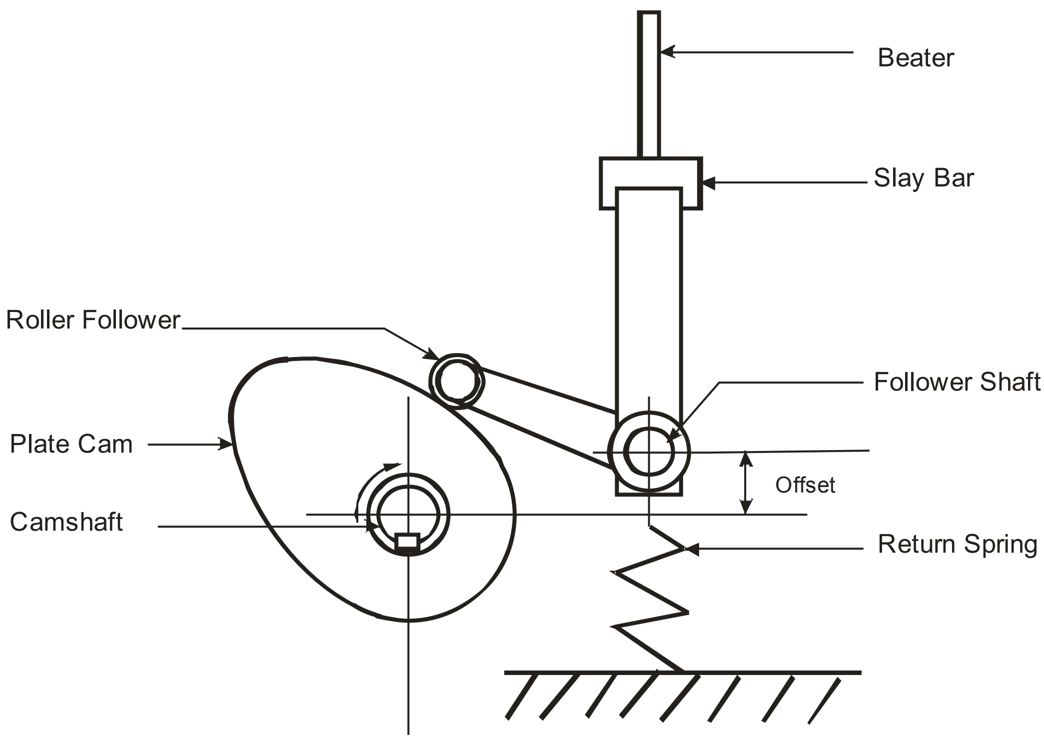


Fig. 1: Schematic diagram of the Model Mechanism.

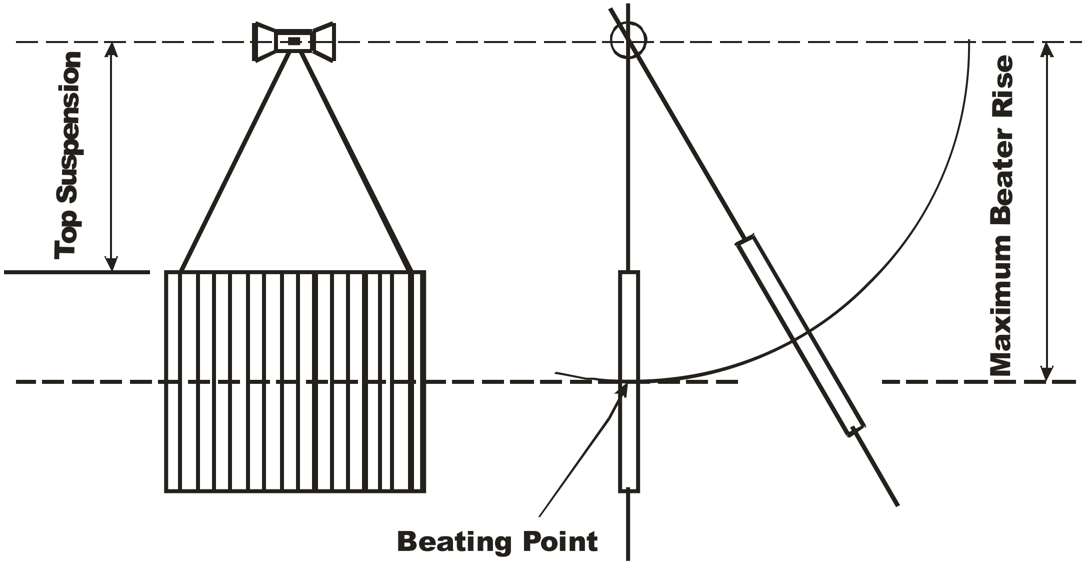


Fig. 2: Schematic View of the Primitive Beater.

The analysis includes the development of the mathematical models required for the design of the various components which make up the mechanism. These include the design of the cam-plate, camshaft, the follower sizing and the drive mechanism.

In the cam-plate the first design consideration is the specification of the motion expected to be exhibited by the plate cam. The motion specification for this study is as developed in Raji, 2000. In the design of cam-plate it is often desired that the minimum size of the cam base circle is achieved in order to save space, reduce weight, and decrease the inertia effect which might arise from the weight of the cam-plate. The resulting minimum base circle reduces the pressure angle of the cam to avoid jamming during operation. The pressure angle is thus desired to be at its minimum in order to achieve the optimum cam size for the design.

Referring to the geometry of the cam–follower combination as shown in Figure 3, the pressure angle is modeled as expressed in equation (1) using the pole method as discussed in Chen, 1982.

$$\alpha = \tan^{-1}\left[\cot(\emptyset + \emptyset_0) - \frac{l\left(1-\frac{d\emptyset}{d\theta}\right)}{c\sin(\emptyset+\emptyset_0)}\right] \quad (1)$$

Where *l* is the length of the roller arm

c is the distance between the cam centre and the follower pivot point

$\emptyset$ is the follower angular displacement

$\emptyset_0$ is the follower angular displacement at start of motion

θ is the cam plate angular displacement

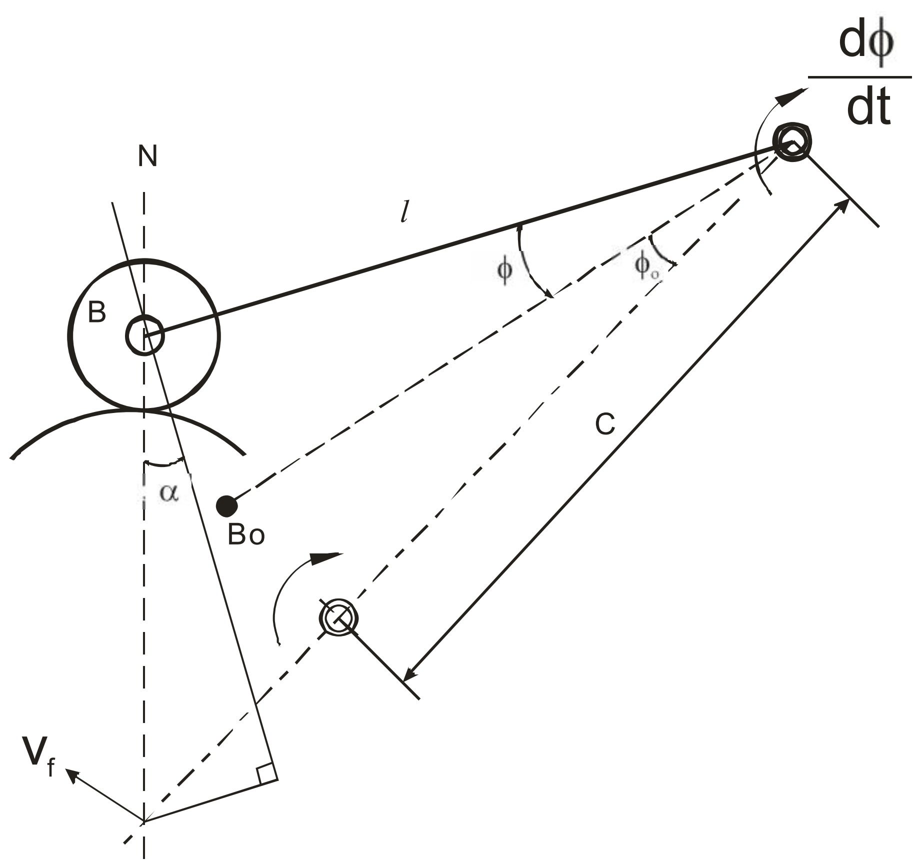


Fig. 3: Geometric Layout for Pressure Angle.

The follower angular displacement at start of motion $\varnothing_0$ could be determined from equation (1) using the FORTRAN code, EDSU to determine the required maximum pressure angle, cam profile and pitch circle radii of curvature, and coordinates of the cam profile and cutter path. It gives direct method for finding the minimum allowable base circle radius based on the criteria that the maximum pressure angle does not exceed a specified value and that profile undercut is avoided. The base circle radius require for the minimum cam size is thus expressed in equation (2)

$$r_0 = \sqrt{l^2 + c^2 - 2lc\cos\varnothing_0} \qquad (2)$$

The least cam size involves specifying the largest possible angular displacement for the follower. This also will minimize the peak follower acceleration with desirable implication for the rotation and follower angular displacement. The velocity of the follower is thus determined as expressed in equation (3).

$$V_f = \frac{c\left(\frac{d\varnothing}{dt}\right)}{\left(1 - \frac{d\varnothing}{dt}\right)} \qquad (3)$$

The follower velocity played an important role in the determination of the impact force required for beating the weft into the yarn to make the fabric. The other critical geometry parameter for the cam design is the radius of curvature which is expressed as given in equation (4) for the swinging roller follower. Chen, 1982.

$$\frac{1}{\rho} = \mu\{1 + \mu[(1 - \varnothing')\varnothing' \sin\alpha - \varnothing'' \cos\alpha]\} \qquad (4)$$

Where $\mu = \frac{\cos\alpha}{c\sin\alpha}$

motion of the follower and to ensure that the compressive contact stresses at the cam follower interface are not excessive.

The force resisted by the cam and roller surface includes the working force and inertia force. The working load which represents the useful work done by the mechanism is obtained as expressed in equation (5).

$$Fi = Ma \quad (5)$$

Where M is the combined mass of the slay bar and beater, and 'a' is the acceleration of the beater as discussed in Raji and Adegbuyi, 2003. Hence the force could be modeled as given in equations (6).

For the rise portion,

$$Fi = M\left[\frac{20}{3\beta^2}(6\tau - 15\tau^2 + 8\tau^3)\right] \quad (6a)$$

For the fall portion,

$$Fi = -M\left[\frac{20}{3\beta^2}(1 - 9\tau^2 + 8\tau^3)\right] \quad (6b)$$

Where τ is the non-dimensional angular ratio for the cam and β is the equivalent cam rotation at maximum rise of the follower. Equations (6) determine the forces required to be generated for the beater to beat the weft into the yarn in order to achieve the required fabric. The turning effect of the follower resulting from the motion of the cam is expressed as;

$$T_f = I \cdot \frac{d\emptyset}{dt} \quad (7)$$

I is the moment of inertia of the follower and T_f is the turning effect on the follower resulting from the motion of the cam and is determined from the expression as in equation (8)

$$T_f = l \times Fi \quad (8)$$

The torque generated by the cam plate is expressed as;

$$T_\theta = y \times Fi \quad (9)$$

The expression for the beater displacement y is discussed in Raji and Adegbuyi, 2003. Thus, for the rise portion,

$$T_\theta = \frac{1}{3}[20\tau^3 - 25\tau^4 + 8\tau^5] \cdot Fi \quad (10)$$

For the return portion,

$$T_\theta = \left[1 - \frac{10}{3}\tau^3 - 5\tau^4 + \frac{8}{3}\tau^5\right] \cdot Fi \quad (11)$$

The power requirement is thus expressed as in equation (12).

$$P_o = \frac{2\pi N T_\theta}{60} \quad (12)$$

N is the speed of the cam plate which should be specified.

The main function of the spring is to maintain contact between the follower and the cam during the motion stroke, especially during the deceleration period. The size of the spring should be as small as possible to minimize the rate of wear and to reduce unnecessary side thrust on the follower, Bounce will occur when the contact force of the system is equal to zero Chen, 1982. The equation of motion for the spring is modeled as expressed in equation (13).

$$F - Mg - k\Delta x = Fi \quad (13)$$

Where F is the impact force, k is the spring constant and Δx is the spring displacement. To prevent bounce, the force F must be larger than zero. Thus, equation (13) results in equation (14)

$$Ma + Mg + k\Delta x > 0 \quad (14)$$

Hence,

$$k > \frac{Mg - Fi}{\Delta x} \quad (15)$$

3.0 FABRICATION

A prototype of the system is eventually built to verify the adequacy of the system. The fabrication process of the proposed mechanism is very simple. It includes the cutting of the cam plate profile from a mild steel of 5 mm thickness. The plate cam is secured on the camshaft also of mild steel and of diameter 25 mm using a woodruff key. The camshaft is driven by a belt-pulley system attached to a variable speed D.C motor. The swinging follower is assembled on the cam and attached to the slay bar on which the beater is assembled. The details of the fabrication process will be presented elsewhere.

4.0 RESULTS AND DISCUSSIONS

Table 1: Force - Cam angular displacement data.

Non-dimensional angular ratio, τ	Impact force, $\square^{2}$Fi/M (Rise portion of beater)	Impact force, $\square^{2}$Fi/M (Return portion of beater)
0	0.000	0.000
0.1	3.053	-3.053
0.2	4.427	-4.427
0.3	4.440	-4.440
0.4	3.413	-3.413
0.5	1.667	-1.667
0.6	-0.480	0.480
0.7	-2.707	2.707
0.8	-4.693	4.693
0.9	-6.120	6.120
1	-6.667	6.667

Table 2: Velocity - Displacement Curve for the Beater Motion (Source; Raji (2000))

Non-dimensional angular ratio, τ	Cam Velocity, βV(τ)
0	0.000
0.1	0.168
0.2	0.555
0.3	1.008
0.4	1.408
0.5	1.667
0.6	1.728
0.7	1.568
0.8	1.195
0.9	0.640
1	0
1.1	-0.648
1.2	-1.195
1.3	-1.568
1.4	-1.728
1.5	-1.667
1.6	-1.408
1.7	-1.008
1.8	-0.555
1.9	-0.168
2	0.000

The performance of the mechanism has also been examined. The response of the beat-up mechanism under the turning effects of the camshaft is investigated and the behavior of the force delivered on the yarn by the beater for the rise and fall motion is examined. The result of the force analysis is presented in Table 1. Figure 4 shows the impact force for the rise and fall portions of the beater motion. The maximum force required to push the weft effectively into the yarn is obtained at the maximum rise of the follower as depicted when the non-dimensional angular displacement is 1. The performance of the mechanism is also measured in terms of fluctuation in the speed of the cam plate as presented in Table 2 and shown in Figure 5. This particular configuration improves the performance of beater operation by increasing the speed of weaving. At maximum impact force the velocity of the beater is modeled to be zero. This is achieved by ensuring that bouncing of the follower is avoided ensuring continuous contact of the follower and cam plate for both rise and return portion of the motion.

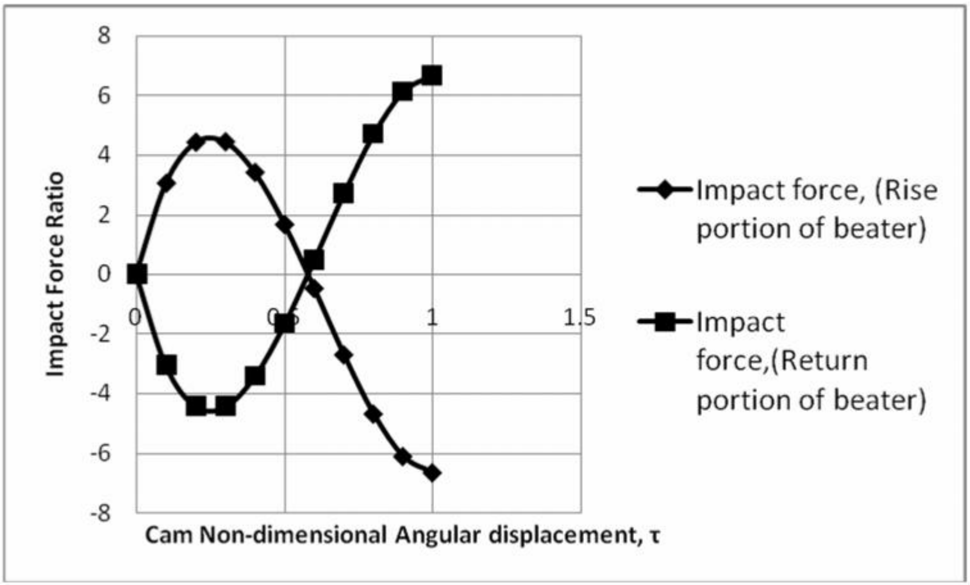
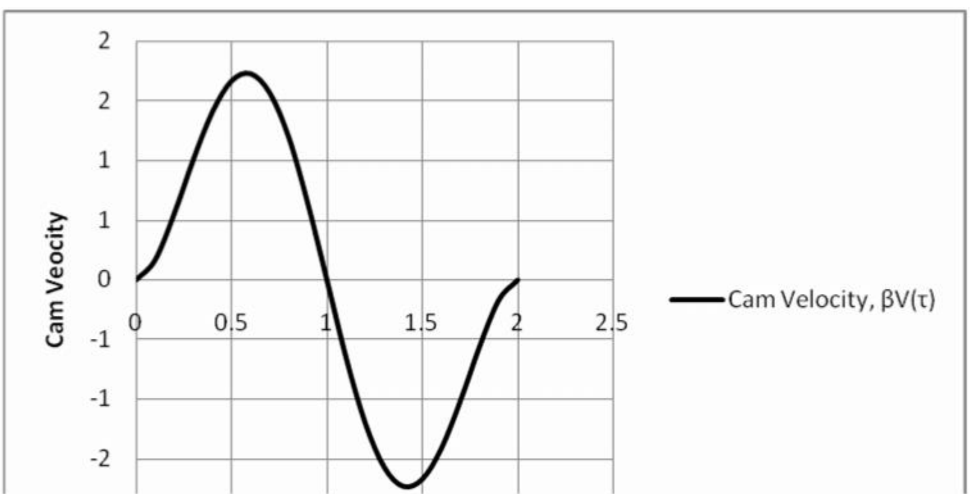


Fig. 4: Plot of the Beater system Impact Force against Cam angular Displacement



5.0 CONCLUSIONS

We have presented a prime mover actuated mechanism design for the beater operation of a narrow weaving loom. This mechanism enables us to control the acceleration of the cam plate at the start and end of motion thereby controlling the impact force of the beater at varying speed of the cam plate. This idea of mechanism tends towards increasing the output and improves the quality of weave from the narrow weaving loom. Additionally, this design prevents the interaction of the process with the actuation part, which enhances its robustness.

REFERENCES

Chen, F.Y., 1982. Mechanics and Design of Cam Mechanisms. Pergamon Press Inc., N.Y. 519.

EDSU International Software, London. N1 6UA Subseries (0171): 490 -5151

Kannapan, S. M and Marshek, K. M., 1990. An algebraic and Predicate Logic Approach to Representation and Reasoning in Machine Design. Mechanism and Machine Theory. 25, (3): 335 – 353.

Krammer, S and Premkumar, P., 1988. A Generalized Expert System Shell for Implementing Mechanical Design Applications; Review, Introduction and Fundamental Concepts, Proceeding of the 1988 ASME Design Automation Conference, American Society of Mechanical engineers.

Raji, N. A., 2000. Motion Specification of the Beating up Mechanism of a Narrow Horizontal Loom. Annals of Engineering Analysis. 1, (2): 28 – 40.

Raji, N. A and Adegbuyi, P. A. O., 2003. Synthesis of Cam – Follower System for the Beater Mechanism of the Narrow Horizontal Loom. Nigerian Journal of Mechanical Engineering. 1, (1): 37 – 48.

Recep Erea, Gulcan Ozkan and Mehmet Karahan., 2005. Comparison of Heald Frame Motion generated by Rotary Dobby and Crank & Cam Shedding Motions. Journal of Fibres and Textile in Eastern Europe. October/December 2005. 13, (4): 78–83.

Sridha Kota and Shean-Juinn Chiou., 1992. Conceptual Design of Mechanism Based on computational Synthesis. Journal of Research in Engineering Design. 4: 75 – 87.

Stanescu, C. D and Samer Florin., 2008. Mechanism which provide Oscillating Motion. Annals of the ORADEA University. VII, (XVII): 603 – 605.

Thompson, T. R., Riley, D. R and Erdman, A. G., 1985. An Expert System Approach to Type Synthesis of Mechanisms. Proceedings of the ASME Computers in Engineering Conference. American Society of Mechanical Engineers, Boston MA, November 1985: 71 – 76.