

Description

This invention relates to a machine module for performing operations on strip material which is intermittently fed through a machine. The disclosed embodiment is in form of a stamping and forming machine for producing electrical terminals or the like.

The disclosed embodiment incorporates strip feeders of the type as claimed in EP—A—0110594, referred to below. A machine module in accordance with the invention can, however, be used with other types or strip feeders.

Strip metal stamping and forming operations are widely used to produce articles such as fasteners or electrical terminals in continuous strip form. The commonly known type of forming apparatus comprises a press having a C-shaped frame member and a progressive die assembly mounted in the press. The die assembly has upper and lower die shoes, the upper shoe being reciprocated towards the lower shoe by a ram in the upper arm of the press which in turn is continuously reciprocated by a crank or eccentric. A feeding mechanism intermittently feeds the strip material through the press and between the upper and lower die shoes which contain a plurality of individual die stations. Each station will contain complementary upper and lower tooling for carrying out the operations which are performed on the material as it is fed through the die assembly, for example, a blanking or profiling operation which may be followed by several punching and forming operations during which individual articles are formed from the strip.

Stamping presses as described above are widely used and have been used ever since continuous stamping became an efficient manufacturing process. The equipment presently used for such operations has been satisfactory in the past and is highly reliable, however, presently used equipment has many shortcomings and disadvantages which must be contended with in present day strip manufacturing operations. For example, many of the parts made by stamping such as electrical terminals or fasteners are quite small and are produced from strip material having a thickness of 0.381 mm (0.015 inches) or less. However, the presses used for the forming operations are relatively massive and would appear to be greatly oversized relative to the scale of the operations being carried out. In fact, relatively massive presses and die shoe assemblies are required because the loading of the parts of the press is eccentric or non-symmetrical and the parts, such as the frame casting, must be enlarged so that they will be able to withstand the eccentric loading during millions of cycles of operation.

Because of the relatively high masses of the parts, presses now used for stamping and forming operations produce a very high level of noise in the work place and increasingly, it is becoming necessary to take steps to reduce the level of noise for reasons of the health of the workers.

The stroke of most presses used for high speed stamping and forming operations is extremely long relative to the nature of the operations being performed on the strip material; in other words, the stroke of the press will frequently have many times the maximum lateral dimension of the part being produced. As a result of the fact that the stroke is unduly long, the inertia developed during each cycle is relatively high and the linear speed of the press ram is very high, particularly if the press is operated at a high speed, say 500 strokes per minute. These factors result in unduly high power requirements for the press and in the imposition of unduly high stresses in the tooling and other parts which are subject to wear. As a result, the maintenance costs for the tooling are increased, particularly if the press is being used to produce precisely dimensioned parts such as electrical terminals.

The present invention is directed to the achievement of an improved machine for performing stamping and forming, or similar, operations on strip material which will overcome the shortcomings of existing stamping presses. The invention is thus directed to the achievement of a machine which has greatly reduced power requirements, which will be extremely quiet as compared with existing stamping and forming machines, which can be operated at extremely high speeds without accompanying excessive wear of the press parts or the forming tooling and which can be "set up" or modified for different parts in a minimum amount of time. The invention is further directed to the achievement of a machine which does not require a conventional progressive die assembly but which is nonetheless capable of carrying out all of the stamping and forming operations which are now carried out by means of progressive dies.

We have disclosed in our prior application EP—110594 citable for novelty purposes under Article 54(4) EPC, a machine module for performing operations on strip material, the machine being of the type comprising a strip feeder for intermittently feeding the strip material in a vertical plane along a strip feed path, an operating zone on the strip feed path, first and second opposed tool holders in the operating zone, the tool holders being on opposite sides of the strip feed path and being reciprocable between retracted positions, in which the tool holders are relatively remote from the strip feed path, and closed positions, in which the tool holders are proximate to the strip feed path, and actuating means for actuating the strip feeder and for reciprocating the tool holders in timed sequence with actuation of the strip feeder so that the tool holders arrive at, and depart from, their closed positions during dwell of the strip, the actuating means comprising a continuously rotatable power shaft; first and second actuator levers, and first and second connecting links, the power shaft extending parallel to, and being spaced from, the strip feed path in the operating zone, the strip feed path and the vertical diametral plane of the

power shaft lying in a common plane; the first and second actuator levers being on opposite sides of the common plane, the actuator levers being pivoted on first and second pivotal axes which extend parallel to the axis of the power shaft; the first and second connecting links being pivotally connected at one end to the first and second actuator levers and being eccentrically coupled at the other end to the power shaft; the first and second actuator levers being coplanar, the first and second actuator levers and the first and second connecting links being symmetrical with respect to the common plane, the strokes of the first and second tool holders being the same, and coupling means are provided for coupling the first and second actuator levers to the tool holders.

The present invention consists in a machine module for performing operations on strip material, the machine module comprising a strip feeder for feeding the strip material intermittently along a strip feed path, an operating zone on the strip feed path, first and second tool holders in the operating zone, the tool holders being aligned with each other on opposite sides of the strip feed path and being movable towards and away from the strip feed path between retracted positions, in which the tool holders are spaced from the feed path, and closed positions, in which the tool holders are substantially against strip material on the strip feed path, and actuating means for actuating the strip feeder and for moving the tool holders between their retracted and closed positions in timed sequence with the strip feeder so that the tool holders are against the strip material during dwell of the strip, the actuating means comprising a continuously rotatable power shaft, first and second actuator levers, and first and second connecting links, the power shaft extending parallel to, and being spaced from, the strip feed path, the actuator levers being on opposite sides of the strip feed path, the connecting links being eccentrically coupled to the power shaft and extending to the actuator levers, the connecting links being pivotally connected to the actuating levers at locations intermediate the ends of the actuating levers, the actuating levers each having a force applying end and a pivoted end, the force applying ends being coupled to the tool holders, the pivoted ends being pivotally mounted on parallel pivotal axes which extend parallel to the main power shaft, the actuator levers being in vertical planes, the connecting links extending horizontally to the power shaft, and the actuator levers are of equal mass and moment of inertia, the connecting links are of equal mass and moment of inertia, and the tool holders are of equal mass and moment of inertia, the strokes of the tool holders being equal whereby the module is dynamically balanced during operation, the module being symmetrical with respect to a plane of symmetry which is defined by the strip feed path and the axis of the power shaft.

The strip feeder is located between the strip

feed path and the power shaft, the first and second actuator levers are coplanar, and the first and second actuator levers and the first and second connecting links are symmetrical with respect to the common plane.

The invention will now be described, by way of example, with reference to the accompanying partly diagrammatic drawings, in which:—

Figure 1 is a perspective view of a machine which is composed of a plurality of machine modules mounted on a bed.

Figure 2 is a cross section taken along the lines 2—2 of Figure 1.

Figure 3 is a perspective view showing one of the machine modules with one of the housing sections removed to reveal the actuating assembly contained in the housing.

Figure 4 is a view similar to Figure 3 but showing the parts of only one of the actuating assemblies and showing the parts exploded from each other.

Figure 5 is a semi-diagrammatic perspective view showing the strip feeders of two adjacent modules with the strip guides exploded from the strip feed path.

Figure 6 is a view taken along the lines 6—6 of Figure 2 and is a side view of a strip feeder.

Figure 7 is a view similar to Figure 6 but showing the strip feeder in cross section.

Figure 8 is a view showing the cylindrical surface of one of the feed screws developed as a flat surface.

Figure 9 is a fragmentary perspective view with parts broken away showing the first feed screw in the machine in which notches are punched in the lower edge of the strip being fed therethrough.

Figure 10 is a perspective view of one side of the tooling assembly of one of the modules.

Figure 11 is a perspective view of the tooling assembly of Figure 10 with the parts exploded from each other.

Figure 12 is a view taken along the lines 12—12 of Figure 10.

Figure 13 is a semi-diagrammatic frontal view of the right-hand side of a module of the apparatus showing only the power shaft, the connecting link, and the actuator lever.

Figure 14 is a diagrammatic representation of the actuator portion of the machine.

Figure 1 shows a machine 2 for performing operations on strip material 4 which is drawn from a reel 6. During passage through the machine, stamping and forming operations, or other operations, are performed on the strip and the processed strip 4' is wound onto a take-up reel 6'. During feeding of the strip, notches 5 are produced in the lower edge 7 thereof as shown in Figure 5 and as will be explained below. The strip will otherwise be modified in accordance with the type of operations carried out; for example, side-by-side contact terminals may be formed from the strip as it passes through the machine.

The machine comprises a plurality of individual machine modules 8 which are mounted on a machine bed 10. The modules may be identical to

each other, excepting for the individual tools which are mounted therein so that a description of one will suffice for all.

Each module 8 comprises a housing assembly 12 composed of two housing parts or sections 14, 16 which are separated by a lower spacer 18 and a pair of upper spacers 20, 22, see Figures 2 and 3. The housing sections 14, 16 are secured to each other by suitable fasteners and are precisely positioned by aligning pins as shown at 24. The housing assembly has an upper surface 26, and laterally facing side surfaces 28, and a base 30 which has dove tails by means of which it is slideably mounted on the machine base 10.

Each module contains a strip feeder assembly 32, tool holder assemblies 34, on each side of the strip feed path, and actuator assemblies 36 on each side of the strip feed path. In the description which follows, the strip feeder assembly 32 will first be described and a tooling assembly will thereafter be described. Alternative types of strip feeders might be used in modules as described below.

The strip feeder is contained in a recess 37 (Figure 3) in the upper surface 26 of the module housing and comprises a pair of spaced part feed screws 38, 40, see Figures 5—10. Each feed screw has a cylindrical surface 42 having a feeding thread 44 thereon which extends from one end 43 of the feed screw to the other end 45. The feeding thread, Figure 8, has a plurality of turns on the surface 42 and each turn has a feeding portion 46 and a dwell portion 48. The feeding portion 46 of each turn extends helically with respect to the axis rotation while the individual dwell portions 48 define planes which extend normally of the axis of rotation. The feeding portion 46 has an acceleration portion and a deceleration portion at its ends so that the strip will be accelerated at the beginning of each feeding cycle and decelerated at the end of the feeding cycle. The dwell portion 48 of each turn is enlarged as shown at 50 at its beginning for purposes of precisely positioning the strip with respect to the tooling assembly.

As shown in Figure 7, the feed screw 38 has an inwardly directed flange 52 which is against the outwardly directed flange 54 on a hollow sleeve 56 which surrounds the continuously rotatable feed shaft 70. Feed screw 38 is precisely positioned on the sleeve 56 by means of aligning pins 58 and secured in position by fasteners 60. The feed screw 40 has an inwardly directed flange 62 which is similarly positioned against a flange 64 on an adaptor 66 which is secured on a reduced diameter end 68 of the sleeve 56. Aligning pins and fasteners locate screw 40 and secure it to the flange as previously explained. The feed screws must be precisely located relative to each other in a rotational sense so that the threads on both of the feed screws will enter the notches 5 in the strip 4.

The feed shaft 70 is splined and the interior of the sleeve 56 has splines as shown at 73. The sleeve 56 is secured to the shaft 70 by bushings 72 which are externally and internally splined. The

entire strip feeder can thus be moved axially along the shaft 70 when adjustment is necessary.

Sleeve 56 extends through a bearing support housing assembly 74 which is disposed in the previously identified recess 37 and which has a base 88 that is above a cover plate 76 that spans the housing sections 14, 16. Ball bearings 78 and raceways are provided between the surface of the sleeve 56 and the interior of the bearing support 74 so that the sleeve and the feed screws 38, 40 will rotate with the shaft 70.

Shaft 70 has a pulley 80 on its end, Figure 5, which is coupled by a belt 82 to a pulley 84 on the main power shaft 86 which is continuously rotated during operation by a motor 87, Figure 1. The main power shaft extends parallel to the feed shaft 70 and the axis of the two shafts define a vertical plane which extends symmetrically through the module as shown in Figure 2.

The strip feeder is mounted in recess 37 between the opposed surfaces 90, 92 of the upper spacers 20, 22 and the upper portions of the housing sections. The mounting is achieved by means of linear adjustable bearings 94, Figure 2, in a manner which permits precise location of the strip during the dwell portion of each cycle as explained.

The strip 4 is guided along the strip feed path and through the modules by means of strip guide assemblies 93, Figure 5 and 10, which are provided adjacent to each of the feed screws 38, 40. These guide assemblies each comprise a pair of complementary blocks 96, 98 having opposed surfaces 100, 102 respectively. The blocks are secured to each other by fasteners and the surface 100 of the block 96 has a ledge 104 which is received in a recess in the other block 98 when the two blocks are against each other. The block 98 has an overhanging ledge 106 on its surface 102 and the two ledges define a slot which guides the strip into, and from, the operating zone of the module; in other words, the zone in which the operation on the strip is carried out.

As best shown by Figure 5, the guide block assemblies 93 are identical to each other and the upstream guide for each strip feeder 32 has a block 98 on the left side of the strip feed path (as viewed in Figure 5) and a block 96 on the right side of the strip feed path so that the ends 108, 110 of the blocks face leftwardly (downstream) as viewed in Figure 5. The downstream guide assembly 93 associated with each strip feeder is adjacent to the feed screw 40 and it has a block 96 on the left-hand side of the strip feed path and a block 98 on the right-hand side of the path with the ends 108, 110 facing upstream. The blocks are secured against a tool holder guide block 116 which is shown in Figure 11 and which is described below.

The strip 4 may be notched prior to its being fed into the machine, however, it is preferably to form the notches 5 in the first strip guide 93' shown in Figure 5 by means of a notching punch 114 which is provided on the feed screw 38' adjacent to, but spaced from, the feed thread on the feed screw

38', see Figure 6. The notching punch 114 cooperates with a notching die 115 which is provided in and insert in 113 block 96' of the guide 93'. The remaining strip guides are identical to each other and do not have a notching die therein. Brushes may be provided as shown in Figure 9 to remove the chips produced by the notching step.

It will be apparent from the foregoing description that during continuous rotation of the shaft 70, the strip will be moved through the machine by each of the strip feeders 32 on each of the modules. The notches will be formed by the upstream feed screw 38' in the first module and during each rotation of the shaft 70, the strip will dwell for a portion of the cycle. During the dwell period, an operation is performed on the strip by the tooling assemblies which are described below.

Referring now to Figures 2 and 10—12, the tooling assemblies 34 on each side of the strip feed path have reciprocating parts which move towards each other and which are mounted in a guide block 116 that is mounted on plates 117 on the upper surface 26 of the housing assembly adjacent to the recess 37. The guide block 116 has laterally facing side surfaces 118 and a passageway 120 which extends between the surfaces 118 for the reciprocating tooling parts. Block 116 has an end surface 122 which faces upstream relative to the direction of strip movement and an end surface 124 which faces downstream. Recesses 126, 128 are provided in the surfaces 122, 124 for the strip guide assemblies 93. A strip guide slot 130 extends through block 116 between the surfaces 122, 124 and is in alignment with the guide slots or passageways in the side assemblies 93. The lower surface of block 116 is covered by cover members 132 on each side of the slot 130, see Figure 2.

The tooling assembly shown in Figure 11 carries a forming tool 133 which is mounted or carried in a tool holder plate 136 and extends through an opening in plate 136. The forward portion of the tool 133 extends freely through an opening 134 in a stripper plate 135. A retaining plate 137 is disposed against the tool holder plate 136 and a plurality of pins 138 extend slideably through aligned openings in the plates 136, 137 and bear against the left-hand surface, as viewed in Figure 11, of the stripper plate 135. At their other ends, pins 138 bear against a disc 139 which is slideably contained in a bore 141 in a slide block 142. A spring 140 is located between the disc 139 and the inner end of the bore 141. Screws 143 extend freely through the slide block 142, through the plate 137, through the tool holder 136, and are threaded into the stripper plate 135. As explained below, the stripper plate is movable relatively leftwardly from the position shown in Figure 12 until it is against the tool holder plate 136 with accompanying compression of the spring 140. The stroke of the assembly shown in Figure 12 is therefore partially represented by the gap 163 between the tool holder plate 136 and the stripper plate 135.

The slide block 142 is secured to a cylindrical slide 144 by means of screws 145 and the cylindrical slide has a projection 146 which bears against a

bearing block 147 which is contained in a bore 148 in a coupling 149. Slide 144 is contained in a cylindrical guide 150 that has a flange 152 by means of which is secured against the surface 118 of the block 116. A spring 151 surrounds the guide 150 and bears against the coupling 149 so that the coupling is biased leftwardly as viewed in Figure 12 by this spring.

Motion is imparted to the reciprocating parts of the tooling assembly by a thrust screw 166 on the actuating mechanism described below. The screw 166 has a spherical end 167 that is received in a recessed thrust disc 153. The disc 153 is held in a spacer 154 in the bore 148 at the left-hand face of the coupling 149. A frangible (capable of being shattered) disc 156 is provided, the disc 156 being between the disc 153 and a ring 155 that has a reduced diameter opening 158. The frangible disc 156 is designed such that it will fracture in the event of a jam in the apparatus and permit movement of the thrust screw 166 without accompanying movement of the reciprocating parts and thereby avoid damage.

Figure 12 shows the positions of the parts of the tooling assembly in its retracted position and during each operating cycle, these parts will move rightwardly until the strip is engaged by the tooling and the operation is performed. The tooling assembly on the right-hand side of the strip feed path will also move inwardly towards the strip in synchronism with the tooling shown in Figure 12. During the operating cycle, the screw 166 will move rightwardly and thereby move all of the parts in the passageway 120 rightwardly until the face 157 of the stripper plate 135 moves into the slot 130 at which time a shoulder 159 on the stripper plate will be against the strip adjacent to its lower edge. The corresponding shoulder of the tooling assembly on the right-hand side of the operating zone will also be against the strip so that the strip will be pinched or held between two surfaces 159. Thereafter, the stripper plate 135 remains stationary and the slide 142 continues to move rightwardly so that the forming tool 133 is moved relatively to the stripper plate 135 and against the strip 4. When the gap 163 is at least partially closed, the inward stroke is completed and the screw 166 thereafter moves leftwardly. Initially, the slide 142 moves leftwardly during the return stroke and thereafter the spring 140 returns the stripper plate 135 to the relative position shown in Figure 12.

As previously mentioned, the stroke of the tooling assembly in Figure 12 is quite small and is represented by the gap 163. However, the tooling assembly on the right-hand side of the center line will also have a stroke of the same magnitude and the total stroke which is effectively available for the forming operation is the sum of the two.

Referring now to Figures 2—4, the actuating assembly comprises the main power shaft 86, connecting links 160, and actuator levers 162. Each tooling assembly 34 has a connecting link and an actuator lever associated therewith.

The upper end 164 of the actuator lever 162 has

the previously identified thrust screw 166 threaded therethrough. This screw can be adjusted to vary the limits of the stroke of the reciprocating parts. Each actuator lever 162 is pivoted at its lower end 168 and is pivoted intermediate its ends at 172 to its associated connecting link 160. As shown in Figure 4, the connecting link is recessed as shown at 170 to receive the actuator lever and has a recess 174 at its inner end 176. The two connecting links 160 are identical to each other and can be mounted on the main power shaft 86 with the inner ends 176 overlapping each other as shown in Figure 3. The inner ends of the connecting links are coupled to the main power shaft by eccentrics 178 which, during each complete rotation of the power shaft, move the connecting links in opposite directions away from and towards each other so that the actuator levers are oscillated in opposite directions towards and away from each other.

As previously mentioned, each module is symmetrical with respect to a plane which would be passed through the centers of the main power shaft 86 and the feed shaft 70. A plane extending through these shafts would also extend through the feed path of the strip as shown best at Figure 2. For best results, and to achieve a high degree of dynamic balance in the apparatus, the two actuator levers are of the same mass and moment of inertia, the two connecting links are of the same mass and moment of inertia, and the parts of the tooling assembly on each side of the center line are similarly balanced.

There are no strict dimensional limitations which must be observed in designing a machine in accordance with the invention, however, the individual modules should have a relatively small eccentricity in the eccentric couplings 178 relative to the dimensions of the connecting links 160 and the actuating levers 162. A brief discussion is presented below of the dimensions of a particular machine in accordance with the invention with reference to Figures 13 and 14. In these figures, the locations of the pivot points and the eccentrics are indicated by letters A—E for convenience in the discussion and some features are exaggerated for purposes of illustration. Particularly, the eccentricities AB and AB' in Figure 14 are greatly exaggerated.

In a preferred embodiment the invention has modules which have an eccentricity AB, AB' of 0.318 mm and has connecting links, AC, AC' which are 165.1 mm in length. The distance between the two pivotal axes 168, 172 of each of the actuator levers DC, DC' is 167.6 mm and the distance from the pivot point D to the tool loading point E is 335.3 mm. The distance AD, AD' (the length of the base link) is about 236 mm. A module 8 having these dimensions and having an eccentric as noted above has a stroke in each tooling block 135 of 1.27 mm so that the total stroke is actually 2.5 mm. This stroke is adequate for many profiling or similar operations which are carried out in the manufacture of sheet metal terminals. The eccentricity can be increased to 2.5

mm to produce a stroke in each tool block of 10.2 mm to yield a total stroke of 20.4 mm. A module having these dimensions will develop a force of about 2,500 kg on the strip.

It can be seen from the foregoing discussion that a module 8 of a machine 2 in accordance with the invention is not massive when compared with conventional presses of the type which are commonly used for performing stamping and forming operations on sheet metal. The individual modules in the machine 2 can moreover have different strokes which would be tailored to the precise operation being carried out in each station.

As mentioned previously, a machine 2 as illustrated above has several advantages as compared with stamping and forming presses of the type currently in use. Some of the significant advantages are reduced power requirements coupled with the capability for operations at high speeds (2,000 to 3,000 or most strokes per minute), quietness in operation, reduced wear and therefore reduced maintenance on the tooling and moving parts of the machine. These advantages stem from several features of the invention which are discussed in general terms below with reference to Figures 13 and 14.

As is mentioned above, each module has a relatively small eccentricity AB, AB', for example, 2.5 mm and which, for many operations, will be as low as 1.27 mm. The base link length AD or AD' is, however, approximately 236 mm and the ratio of the eccentricity to the base link length AB/AD will always be an extremely low number, much less than unity. If for example, the eccentricity is 1.27 mm, the AB/AD ratio is approximately .005. This condition results in angular velocities and accelerations in the connecting links 160 and the actuator levers 162 which are nearly sinusoidal. Additionally, these velocities and accelerations are relatively low even when the machine is operated at high RPM. As a result, the inertial forces in the machine are minimized and the power requirements are thereby reduced. A further beneficial result is that the linear velocities and accelerations in the tooling blocks 135 are quite low relative to the rotational speed of the shaft 86 so that tool wear and wear on the moving parts of the tooling assemblies is minimized.

The wear of machine parts which move over each other is caused by abrasion and/or erosion. Abrasion wear is the type of wear caused by the mechanical abrading effect of the parts on each other while erosion results when the parts move at very high speeds and a high temperature is developed in an extremely narrow zone at the interface. Erosion wear can be explained in terms of contact physics as related to a phase change in the localized zone and removal of material while it is in the liquid phase. Erosion wear is greatly reduced or eliminated in a machine in accordance with the invention by virtue of the fact that the linear speeds of the parts, as compared with a conventional stamping press, are greatly reduced.

The very small eccentricities AB, AB' coupled with the dimensions of the connecting links 160 and the actuators levers 162 produce an extremely high mechanical advantage at the tool loading points E in Figure 14. This feature again results in low torque requirements in the shaft 86 and therefore reduced power requirements for the machine as a whole.

Referring to Figure 14, another important feature is that the angles BCD and B'C'D' are always relatively close to 90° (these angles being highly exaggerated in Figure 14 for purposes of illustration). These angles BCD, B'C'D' are referred to as the power transmission angles in this type of mechanism and are important in the determination of the efficiency of the mechanism, particularly as regards the bearing loads developed during operation and the portion of the thrust in the connecting links which is transmitted to the tool loading points E, E'. It is recognized that the closer the transmission angles are to 90°, the smaller the vertical force component in the system and the greater the horizontal force component as viewed in Figure 14. The horizontal force component is the useful component which is transmitted to the tool loading point E and is therefore available for performing work on the strip material being fed through the machine. The vertical force component, on the other hand, is in effect lost and must be contained by bearing members and static structural members in the machine. The fact that the transmission angles are always very close to 90° contributes to the low power requirements in that the power supplied to the shaft 86 is effectively utilized and also contributes to the fact that the structural components need not be designed to contain an excessive and useless vertical force component; in other words, this feature contributes to the compactness of the apparatus.

Another feature of the apparatus is that the connecting links 160 are placed in tension during application of the load to the strip material rather than in compression as in a conventional stamping press. The avoidance of compression loading in these members is advantageous in that tension loading is a much more efficient method of loading a machine element than compression loading. While most materials have very good compressive strengths, failure of machine parts as a result of compressive loads must be anticipated as a result of buckling rather than simple compressive failure of the metal. To avoid buckling, a part which is stressed in compression must be made relatively massive and bulky. The fact that the connection links are stressed in tension therefore permits them to be of smaller mass which in turn contributes to the lower power requirements of the apparatus.

Many of the features discussed above contribute to the relative quietness which characterizes a stamping machine incorporating a machine module in accordance with the invention. Much of the sound produced during operation of a conventional stamping press is a result of the impact

ing of the moving parts, particularly in the dies. In a conventional press, the linear speeds of the die parts are considerably higher, other things being equal, than are reached in a machine of the present invention designed in accordance with good design practice. The high linear speeds and the higher masses involved in conventional stamping machines result in a noise level which is often objectionable to the point of being an industrial hazard. A dramatic reduction in the noise level is achieved with the practice with present invention as a result of the low masses of the parts and the lower linear speeds notwithstanding the relatively high operating speed of the machine in terms of revolutions per minute.

The foregoing discussion treats only briefly, and in a qualitative manner, some advantages of the invention. A more rigorous consideration of the operating principles will reveal further benefits.

Claims

1. A machine module (8) for performing operations on strip material (4), the machine module (8) comprising a strip feeder (32) for feeding the strip material (4) intermittently along a strip feed path, an operating zone (12) on the strip feed path, first and second tool holders (136) in the operating zone, the tool holders (136) being aligned with each other on opposite sides of the strip feed path and being movable towards and away from the strip feed path between retracted positions, in which the tool holders are spaced from the feed path, and closed positions, in which the tool holders are substantially against strip material (4) on the strip feed path, and actuating means for actuating the strip feeder (32) and for moving the tool holders (136) between their retracted and closed positions in timed sequence with the strip feeder so that the tool holders are against the strip material (4) during dwell of the strip, the actuating means comprising a continuously rotatable power shaft (86), first and second actuator levers (162), and first and second connecting links (160), the power shaft (86) extending parallel to, and being spaced from, the strip feed path, the actuator levers (162) being on opposite sides of the strip feed path, the connecting links (160) being eccentrically coupled to the power shaft (86) and extending to the actuator levers, the connecting links being pivotally connected (172) to the actuating levers (162) at locations intermediate the ends (164, 168) of the actuating levers, the actuating levers (162) each having a force applying end (164) and a pivoted end (168), the force applying ends (164) being coupled to the tool holders (136), the pivoted ends (168) being pivotally mounted on parallel pivotal axes which extend parallel to the main power shaft (86), the actuator levers (162) being in vertical planes, the connecting links (160) extending horizontally to the power shaft (86), and the actuator levers (162) are of equal mass and moment of inertia, the connecting links (160) are



The very small eccentricities AB, AB' coupled with the dimensions of the connecting links 160 and the actuators levers 162 produce an extremely high mechanical advantage at the tool loading points E in Figure 14. This feature again results in low torque requirements in the shaft 86 and therefore reduced power requirements for the machine as a whole.

Referring to Figure 14, another important feature is that the angles BCD and B'C'D' are always relatively close to 90° (these angles being highly exaggerated in Figure 14 for purposes of illustration). These angles BCD, B'C'D' are referred to as the power transmission angles in this type of mechanism and are important in the determination of the efficiency of the mechanism, particularly as regards the bearing loads developed during operation and the portion of the thrust in the connecting links which is transmitted to the tool loading points E, E'. It is recognized that the closer the transmission angles are to 90°, the smaller the vertical force component in the system and the greater the horizontal force component as viewed in Figure 14. The horizontal force component is the useful component which is transmitted to the tool loading point E and is therefore available for performing work on the strip material being fed through the machine. The vertical force component, on the other hand, is in effect lost and must be contained by bearing members and static structural members in the machine. The fact that the transmission angles are always very close to 90° contributes to the low power requirements in that the power supplied to the shaft 86 is effectively utilized and also contributes to the fact that the structural components need not be designed to contain an excessive and useless vertical force component; in other words, this feature contributes to the compactness of the apparatus.

Another feature of the apparatus is that the connecting links 160 are placed in tension during application of the load to the strip material rather than in compression as in a conventional stamping press. The avoidance of compression loading in these members is advantageous in that tension loading is a much more efficient method of loading a machine element than compression loading. While most materials have very good compressive strengths, failure of machine parts as a result of compressive loads must be anticipated as a result of buckling rather than simple compressive failure of the metal. To avoid buckling, a part which is stressed in compression must be made relatively massive and bulky. The fact that the connection links are stressed in tension therefore permits them to be of smaller mass which in turn contributes to the lower power requirements of the apparatus.

Many of the features discussed above contribute to the relative quietness which characterizes a stamping machine incorporating a machine module in accordance with the invention. Much of the sound produced during operation of a conventional stamping press is a result of the impact-

ing of the moving parts, particularly in the dies. In a conventional press, the linear speeds of the die parts are considerably higher, other things being equal, than are reached in a machine of the present invention designed in accordance with good design practice. The high linear speeds and the higher masses involved in conventional stamping machines result in a noise level which is often objectionable to the point of being an industrial hazard. A dramatic reduction in the noise level is achieved with the practice with present invention as a result of the low masses of the parts and the lower linear speeds notwithstanding the relatively high operating speed of the machine in terms of revolutions per minute.

The foregoing discussion treats only briefly, and in a qualitative manner, some advantages of the invention. A more rigorous consideration of the operating principles will reveal further benefits.

Claims

1. A machine module (8) for performing operations on strip material (4), the machine module (8) comprising a strip feeder (32) for feeding the strip material (4) intermittently along a strip feed path, an operating zone (12) on the strip feed path, first and second tool holders (136) in the operating zone, the tool holders (136) being aligned with each other on opposite sides of the strip feed path and being movable towards and away from the strip feed path between retracted positions, in which the tool holders are spaced from the feed path, and closed positions, in which the tool holders are substantially against strip material (4) on the strip feed path, and actuating means for actuating the strip feeder (32) and for moving the tool holders (136) between their retracted and closed positions in timed sequence with the strip feeder so that the tool holders are against the strip material (4) during dwell of the strip, the actuating means comprising a continuously rotatable power shaft (86), first and second actuator levers (162), and first and second connecting links (160), the power shaft (86) extending parallel to, and being spaced from, the strip feed path, the actuator levers (162) being on opposite sides of the strip feed path, the connecting links (160) being eccentrically coupled to the power shaft (86) and extending to the actuator levers, the connecting links being pivotally connected (172) to the actuating levers (162) at locations intermediate the ends (164, 168) of the actuating levers, the actuating levers (162) each having a force applying end (164) and a pivoted end (168), the force applying ends (164) being coupled to the tool holders (136), the pivoted ends (168) being pivotally mounted on parallel pivotal axes which extend parallel to the main power shaft (86), the actuator levers (162) being in vertical planes, the connecting links (160) extending horizontally to the power shaft (86), and the actuator levers (162) are of equal mass and moment of inertia, the connecting links (160) are



of equal mass and moment of inertia, and the tool holders (136) are of equal mass and moment of inertia, the strokes of the tool holders being equal whereby the module (8) is dynamically balanced during operation, the module (8) being symmetrical with respect to a plane of symmetry which is defined by the strip feed path and the axis of the power shaft (86).

2. A machine module as set forth in claim 1, characterized in that the second actuator levers (162) extend substantially normally of the connecting links (160) and the power transmission angles are substantially 90°.

3. A machine module as set forth in claim 1 or 2, characterized in that the first and second connecting links (160) are eccentrically coupled to the power shaft (86) by means of eccentrics (178) having an eccentricity within the range of 0.318 to 2.5 mm, the distance between the eccentric couplings and the first and second pivotal axes (168) being substantially 236 mm, the lengths of the connecting links (160) being about 165 mm.

4. A machine module as set forth in claim 3, characterized in that said eccentricity is within the range of 1.27 to 2.5 mm.

5. A machine module (8) as set forth in claim 1, characterized in that the actuator levers (162) are coplanar, the actuator levers and the connecting links (160) being symmetrical with respect to a vertical axis which extends through the power shaft (86).

6. A machine module (8) as set forth in claim 1 characterized in that the module is adjustably mounted on a machine bed (10) for movement parallel to the axis of the power shaft (86), the machine bed (10) having at least one additional module (8) adjustably mounted thereon, the additional module (8) being similar to the original module (8), the power shaft (86) and the strip feed path extending through the additional module (8).

Patentansprüche

1. Maschinenmodul (8) zur Ausführung von Arbeitsvorgängen an Bandmaterial (4), wobei der Maschinenmodul (8) aufweist: einen Bandförderer (32) zum intermittierenden Befördern des Bandmaterials (4) entlang einer Bandförderbahn, eine Bearbeitungszone (12) auf der Bandförderbahn, einen ersten und einen zweiten Werkzeughalter (136) in der Bearbeitungszone, wobei die Werkzeughalter (136) auf einander gegenüberliegenden Seiten der Bandförderbahn miteinander ausgerichtet sind und in Richtung auf die Bandförderbahn zu sowie von dieser weg zwischen einer zurückgezogenen Position, in der die Werkzeughalter von der Förderbahn beabstandet sind, und einer geschlossenen Position, in der die Werkzeughalter im wesentlichen an dem auf der Bandförderbahn befindlichen Bandmaterial (4) anliegen, bewegbar sind, sowie eine Betätigungseinrichtung zum Betätigen des Bandförderers (32) und zum Bewegen der Werkzeughalter (136) zwischen ihrer zurückgezogenen und ihrer geschlossenen Position in zeitlich gesteuerter

Abfolge mit dem Bandförderer in einer derartigen Weise, daß die Werkzeughalter während einer Verweilperiode des Bands an dem Bandmaterial (4) anliegen, wobei die Betätigungseinrichtung eine kontinuierlich drehbare Antriebswelle (86), einen ersten und einen zweiten Betätigungshebel (162) und ein erstes und ein zweites Verbindungsglied (160) aufweist, wobei sich die Antriebswelle (86) parallel zu der Bandförderbahn erstreckt und von dieser beabstandet ist, die Betätigungshebel (162) auf einander gegenüberliegenden Seiten der Bandförderbahn angeordnet sind und die Verbindungsglieder (160) mit der Antriebswelle (86) exzentrisch gekoppelt sind und sich zu den Betätigungshebeln erstrecken, wobei die Verbindungsglieder an Stellen zwischen den Enden (164, 166) der Betätigungshebel schwenkbar mit den Betätigungshebeln (162) verbunden sind und die Betätigungshebel (162) jeweils ein Kraftbeaufschlagungsende (164) und ein angelenktes Ende (168) aufweisen, wobei die Kraftbeaufschlagungsenden (164) mit den Werkzeughaltern (136) gekoppelt sind und die angelenkten Enden (168) auf parallelen Schwenkachsen schwenkbar montiert sind, die sich parallel zu der Hauptantriebswelle (86) erstrecken, wobei sich die Betätigungshebel (162) in vertikalen Ebenen befinden und sich die Verbindungsglieder (160) horizontal zu der Antriebswelle (86) erstrecken, und wobei die Betätigungshebel (162) dieselbe Masse und dasselbe Trägheitsmoment aufweisen, die Verbindungsglieder (160) dieselbe Masse und dasselbe Trägheitsmoment aufweisen und die Werkzeughalter (136) dieselbe Masse und dasselbe Trägheitsmoment aufweisen und die Hübe der Werkzeughalter miteinander identisch sind, wodurch der Modul (8) während des Betriebs in dynamischem Gleichgewicht ist, und wobei der Modul (8) symmetrisch in bezug auf eine Symmetrieebene ist, die durch die Bandförderbahn und die Achse der Antriebswelle (86) definiert ist.

2. Maschinenmodul nach Anspruch 1, dadurch gekennzeichnet, daß sich die zweiten Betätigungshebel (162) im wesentlichen senkrecht zu den Verbindungsgliedern (160) erstrecken und die Kraftübertragungswinkel im wesentlichen 90° betragen.

3. Maschinenmodul nach Anspruch 1 oder 2, dadurch gekennzeichnet, daß das erste und das zweite Verbindungsglied (160) mit der Antriebswelle (86) mittels Exzentern (178) exzentrisch gekoppelt sind, die eine Exzentrizität im Bereich von 0,318 bis 2,5 mm aufweisen, wobei die Entfernung zwischen den exzentrischen Kopplungen und der ersten und der zweiten Schwenkachse (168) im wesentlichen 236 mm beträgt und die Längen der Verbindungsglieder (160) ca. 165 mm betragen.

4. Maschinenmodul nach Anspruch 3, dadurch gekennzeichnet, daß die Exzentrizität im Bereich von 1,27 bis 2,5 mm liegt.

5. Maschinenmodul (8) nach Anspruch 1, dadurch gekennzeichnet, daß die Betätigungshebel (162) koplanar sind, und die Betätigungshebel und die Verbindungsglieder (160) in bezug auf

eine durch die Antriebswelle (86) verlaufende vertikale Achse symmetrisch sind.

6. Maschinenmodul (8) nach Anspruch 1, dadurch gekennzeichnet, daß der Modul auf einem Maschinenbett (10) derart verstellbar montiert ist, daß er sich parallel zu der Achse der Antriebswelle (86) bewegen läßt, daß auf dem Maschinenbett (10) wenigstens ein zusätzlicher Modul (8) verstellbar montiert ist, daß der zusätzliche Modul (8) dem ursprünglichen Modul (8) gleichartig ausgebildet ist, und daß sich die Antriebswelle (86) und die Bandförderbahn durch den zusätzlichen Modul (8) erstrecken.

Revendications

1. Module (8) de machine pour effectuer des opérations sur une matière en bande (4), le module (8) de machine comprenant un dispositif (32) d'avance de bande destiné à faire avancer la matière en bande (4) par intermittence le long d'un trajet d'avancer de bande, une zone (12) de travail sur le trajet d'avance de bande, des premier et second porte-outils (136) dans la zone de travail, les porte-outils (136) étant alignés entre eux sur des côtés opposés du trajet d'avance de bande et pouvant être rapprochés et éloignés du trajet d'avance de bande entre des positions rétractées, dans lesquelles les porte-outils sont espacés du trajet d'avance, et des positions fermées, dans lesquelles les porte-outils sont sensiblement contre la matière en bande (4) sur le trajet d'avance de bande, et des moyens d'actionnement destinés à actionner le dispositif (32) d'avance de bande et à déplacer les porte-outils (136) entre les positions rétractées et fermées, en une séquence synchronisée avec le dispositif d'avance de bande, afin que les porte-outils soient appliqués contre la matière en bande (4) durant l'arrêt de la bande, les moyens d'actionnement comprenant un arbre moteur (86) pouvant tourner en continue, des premier et second leviers actionneurs (162), et des première et seconde biellettes de liaison (160), l'arbre moteur (86) s'étendant parallèlement au trajet d'avance de bande et étant espacé de celui-ci, les leviers actionneurs (162) étant situés sur des côtés opposés du trajet d'avance de bande, les biellettes de liaison (160) étant accouplées de façon excentrique à l'arbre moteur (86) et s'étendant jusqu'aux leviers actionneurs, les biellettes de liaison étant articulées (172) sur les leviers d'actionnement (162) en des points situés entre les extrémités (164, 168) des leviers d'actionnement, les leviers d'actionnement (162) ayant chacun une extrémité (164) d'application de force et une

extrémité pivotante (168), les extrémités (164) d'application de force étant accouplées aux porte-outils (136), les extrémités articulées (168) étant montées de façon pivotante sur des axes parallèles de pivotement qui s'étendent parallèlement à l'arbre moteur principal (86), les leviers actionneurs (162) étant dans des plans verticaux, les biellettes de liaison (160) s'étendant horizontalement jusqu'à l'arbre moteur (86), et les leviers actionneurs (162) étant de masses égales et de moments d'inertie égaux, les biellettes de liaison (160) étant de masses égales et de moments d'inertie égaux et les porte-outils (136) étant de masses égales et de moments d'inertie égaux, les courses des porte-outils étant égales de manière que le module (8) soit équilibré dynamiquement en cours de fonctionnement, le module (8) étant symétrique par rapport à un plan de symétrie qui est défini par le trajet d'avance de bande et l'axe de l'arbre moteur (86).

2. Module de machine selon la revendication 1, caractérisé en ce que les seconds leviers actionneurs (162) s'étendent sensiblement perpendiculairement aux biellettes de liaison (160) et les angles de transmission de puissance sont sensiblement de 90°.

3. Module de machine selon la revendication 1 ou 2, caractérisé en ce que les première et seconde biellettes de liaison (160) sont accouplées de façon excentrique à l'arbre moteur (86) au moyen d'excentriques (178) ayant une excentricité dans l'intervalle de 0,318 à 2,5 mm la distance entre les accouplements à excentriques et les premier et second axes de pivotement (168) étant sensiblement de 236 mm les longueurs des biellettes de liaison (160) étant d'environ 165 mm.

4. Module de machine selon la revendication 3, caractérisé en ce que ladite excentricité est comprise dans l'intervalle de 1,27 à 2,5 mm.

5. Module (8) de machine selon la revendication 1, caractérisé en ce que les leviers actionneurs (162) sont coplanaires, les leviers actionneurs et les biellettes de liaison (160) étant symétriques par rapport à une axe vertical qui passe par l'arbre moteur (86).

6. Module (8) de machine selon la revendication 1, caractérisé en ce que le module est monté de façon réglable sur un bâti (10) de machine de façon à se déplacer parallèlement à l'axe de l'arbre moteur (86), le bâti (10) de la machine comportant au moins un module supplémentaire (8) monté de façon réglable sur lui, le module supplémentaire (8) étant similaire au module initial (8), l'arbre moteur (86) et le trajet d'avance de bande s'étendant à travers le module supplémentaire (8).

