# Charles Babbage's Difference Engine No. 2
# Technical Description

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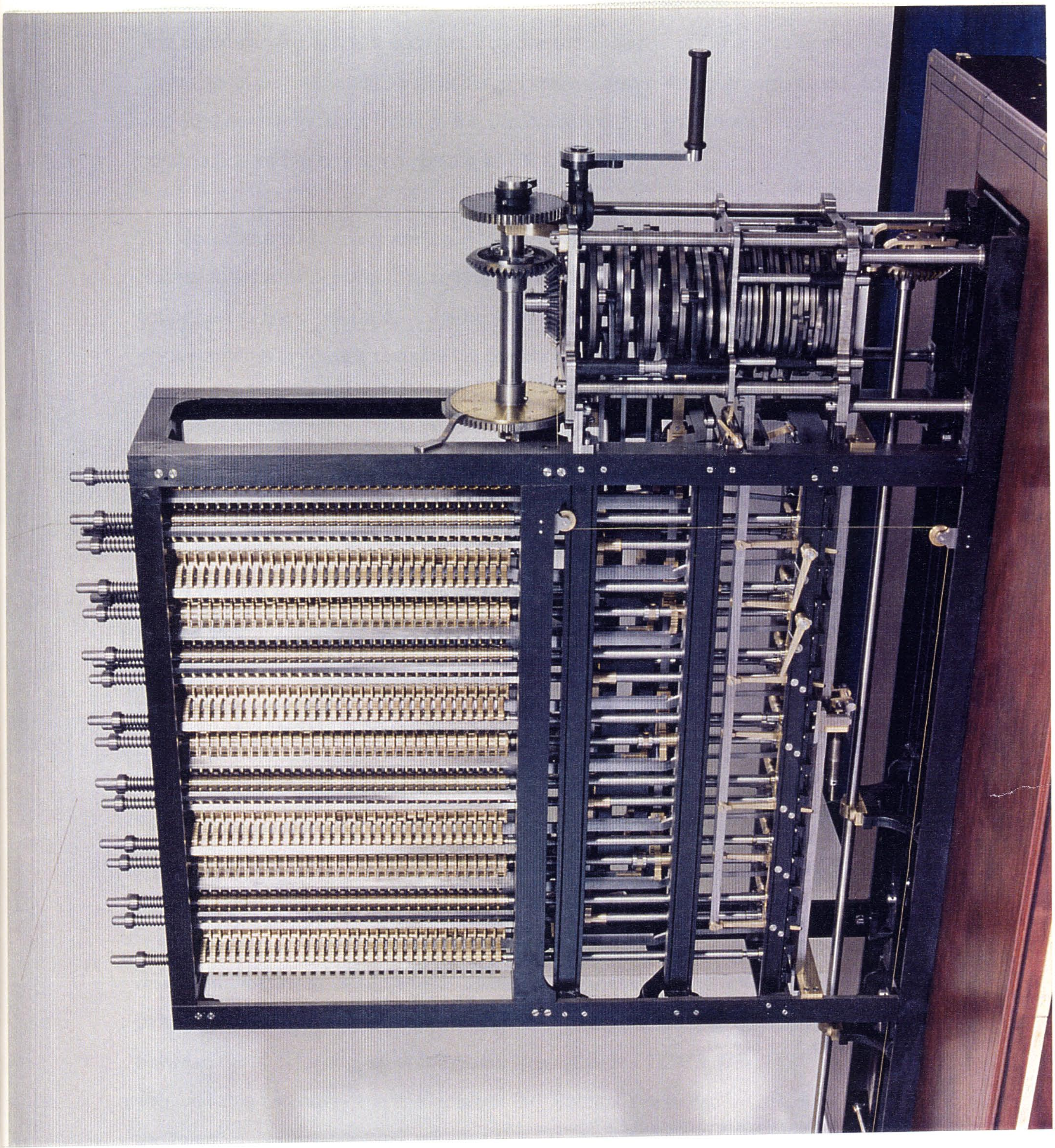
# Contents



## Preface

The accompanying account provides a technical description of Difference Engine No. 2. It consists of an interpretation of Babbage's original design drawings and records the design and engineering decisions made in the preparation of modern piece-part drawings for the construction of the engine completed in 1991.

The documentation project started in January 1993. The account is the product of a series of debriefing sessions between the author and Reg Crick, the engineer who played the major part in the production of the new drawings and in the construction of the engine. The documentation process is described in the Foreword to the 'Work in Progress' report (included below) issued in February last year.

This first draft is issued to mark the completion of the debriefing phase of the documentation process. Since the work will now be intermitted it is also issued as a form of security in the event that the work is not resumed. The contents of the *Work in Progress report (February 1995)* remains unchanged. However, two sections have been added to the account during the last year. The overall account runs to 58,000 words. These sections complete the description of the stereotype table and add a final section covering issues that arose during the physical assembly.

The account is intended as an interpretative companion to the set of twenty original full-sized design drawings and twelve tracings for Difference Engine No. 2, produced by Babbage, or under his instruction, between 1847 and 1849. Babbage made liberal use of his Mechanical Notation in the depiction of the engine in these drawings. The Notation is an elaboarte system of symbols devised by Babbage to identify parts and describe their operation. It consists of alphabetic characters of various kinds which identify fixed and moving parts, and numerical superscripts and subscripts which indicate the operational relationship between parts. The original drawings are large (typically 1000 x 600 mm) and well able to convey the fine detail of the Notational annotations. Reduced copies of the twenty main drawings are provided here in an Appendix. Most, though not all of the alphabetic annotations are legible in these copies but the notational super- and sub-scripts have inevitably suffered in the reduction. The textual account relies on the Notation to identify parts but in general does not rely on it for the description of the operation of mechanisms. Despite the diminished detail of the reduced copies the account that follows does provide a free–standing description of the engine without reference to the full-sized original. However, for detailed study of the Notation and its use in Babbage's description of the engine three is no substitute for reference to the full-sized primary sources.

Some progress was made on the Mechanical Notation during the last year and this helped to resolve a major issue in the design of the stereotype table. However, current understanding of the Notation is still in complete and its study is a research task that remains outstanding.

Doron Swade
15 March 1996
9 September 1996

**Work in Progress (February 1995)**

*This account documents the construction of Difference Engine No. 2 taking the original set of Babbage's drawings as an uninterpreted datum. The purpose of the account is to place on record the detailed interpretation of a Babbage engine design. It also serves to provide a technical record of the process by which a project, arrested in 1849, was resumed and culminated in the completion, in 1991, of an operational machine.*

*Foreword*

The accompanying account describes Charles Babbage's Difference Engine No. 2. The machine was built by the Science Museum to original designs dating from between 1847 and 1849. The engine was completed in 1991, the bicentennial year of Babbage's birth. The set of original design drawings consists of twenty main views and a small number of derivative tracings. The set is believed to be complete i.e. there is no evidence of drawings being missing. The drawings collectively constitute a comprehensive operational description of the engine: they are sufficiently detailed to describe the shape and nominal size of individual parts, their physical interrelationship and their intended function. However, for all the richness they contain the drawings are not 'working drawings' i.e. they are not sufficiently detailed to provide the full specification necessary for the manufacture of parts. No information is provided in the originals as to choice of materials, methods of manufacture, requisite precision, or finish. In this respect they provide a schematic description, comprehensive in itself, but insufficiently detailed to serve directly as a specification for manufacture.

The drawings are deficient in other respects: they contain dimensioning inconsistencies (the same parts are shown differently sized in different views); there are incompletenesses in the design, instances of omitted devices, as well as redundant assemblies. The drawings also contain design errors one of which is central to the fundamental operation of the engine. None of these deficiencies compromises the validity of the basic logic, design or intended function of the engine. Babbage made no practical attempt to construct the engine and the deficiencies for the most part represent the gap between an advanced design arrested in an incomplete stage of engineering development, and the final working mechanical entity. The gap, in short, is one of engineering completeness rather than logical or operational principle. The drawings constitute the only original account of the engine. Babbage provides almost no other explanation, textual description or justification for the design. The drawings are therefore a free-standing source from which his intentions are to be decoded and reconstructed.

The manufacture of parts required fully specified working drawings. These include detailed piece-part drawings, parts lists specifying quantities of individual components, layouts and general assembly drawings detailing the physical interrelation of assembled parts. The production of these drawings required detailed interpretation of the originals to establish the purpose and function of parts and subassemblies. This interpretation then informed the provision of supplementary data, otherwise missing from the originals, but essential for manufacture. Composition analysis of contemporary gunmetal informed the choice and grade of modern bronze; which parts where to be made from bronze, cast iron or steel, methods of manufacture, finish and tolerancing were informed by expert curatorial advice based on detailed knowledge of nineteenth-century practice. The guiding principle throughout was authenticity, and care was taken to ensure that no part was made with greater precision than is known from measurement to have been deliverable in Babbage's time. Dimensioning inconsistencies were resolved, incompletenesses in the design remedied, modifications to correct design errors specified and omissions and redundancies catered for. These researches and deliberations were embodied in a set of some fifty working drawings which completely specify the engine. During the design and build Reg Crick, the senior engineer responsible for the production of the modern drawings and for the supervision
of the construction, kept an Engineer's Log (The Red Book) which is an episodic diary of notable issues. Apart from this we were as guilty as Babbage during the design and build phases of the project in providing little descriptive account of the technical issues underpinning the design. The attached account is intended to remedy this, at least in part.

The documentation process started in January 1993 and has continued fairly consistently since then. The account is based on debriefing sessions between myself and Reg Crick. During the debriefing sessions the interpretation of the original drawings is retraced and reexamined so as to reconstruct the supposed intention and function of the various devices. Deficiencies, where they occur, are identified, and the means and justification by which they were resolved described. The Engineer's Log (The Red Book) is used as a checklist of notable issues. My role in these sessions progresses from disciple, to inquisitor, and finally, to that of scribe. The debriefing sessions are recorded in handwritten summary session notes by me. These are then taken away and used in solitude as the basis of an expanded account written usually on the same day or in the days immediately following. In the next session Reg Crick would proof read the text. Secondary questions raised during the drafting process, as well as amendments to the text from the proofing process would be discussed, clarified and resolved. The account would then be redrafted accordingly and re-read at the start of the following session. The pages attached are the outcome of the process to date.

There are two flagged incompletenesses in the account. These are provisional omissions which require curatorial research outside Reg Crick's or my immediate knowledge. One of the topics is the use of Babbage's Mechanical Notation. Despite the central importance attached by Babbage to the Notation as an interpretative and descriptive tool, we did not use it to decode the drawings nor in the construal of Babbage's intentions. While I have used the Notation in the later sections as a short-hand to identify parts, this is the main use to which it has so far been put in the written record. The account will need to be reviewed in the light of a more informed understanding of the Notation to verify and confirm interpretative decisions taken without its aid. An investigation of the notational conventions devised by Babbage is a necessary curatorial research task but is being held in abeyance until the technical account, which relies on Reg Crick's unique experience and knowledge, is complete. The second issue is the historical account of a major design error in the addition mechanism. This requires reconstructing the sequence of interpretative exchanges between Prof. Allan Bromley and the engine team. Retracing the trajectory of understanding here is not essential to the debriefing process. This too has been postponed for the meanwhile but will need to be returned to for completeness.

The engine depicted in the original design drawings consists of a calculating section to which is attached a printing and stereotyping apparatus. Only the calculating section has so far been built. The description of the calculating section is retrospective i.e. this part of the engine existed as a *fait accomplis* at the time of writing. The account is therefore 'descriptive' in the literal sense (*de* – down, *scribo* – I write) and my role here is largely that of a scribe — retracing, recovering, articulating and recording, via a process of understanding, a set of considerations and events already enacted. The account, however, also covers the printing and stereotyping section which remains unbuilt but for which working drawings were produced in the interval since the bicentennial year, funded by independent sponsorship from the United States. The relationship between the process of documentation and the process of design is in this case different.

The printing and stereotyping section is integral to the concept of the machine and forms an essential part of the engine's control system. The apparatus is at least as complex as the calculating section, and calls for an additional 4,000 parts - about the same number as required for the calculating section already built - but with substantially less repetition of similar parts. Documenting the printing and stereotyping apparatus involved the familiar process of re–examining the existing interpretation of the original design intention, and retracing the decisions made in designing the incompletely designed parts of the mechanism. The opportunity for curatorial scrutiny was greater here than circumstances allowed during the earlier build of the calculating section, and since there still existed the possibility to re–interpret the design, the curatorial responsibility was correspondingly greater. The process of reviewing the design substantially altered our understanding of the printing mechanism and revealed features of startling subtlety formerly overlooked. The design was subsequently altered to accommodate this partial re-interpretation. I cite this to illustrate how, in the case of the printing apparatus, the documentation process and the design process directly interacted and breached what had until then been separate activities.

The account consists of seven sections which collectively run to 45,460 words. A total of 300 hours has been spent on debriefing and writing up since January 1993 (192 hours in 1993, 96 hours in 1994, and 12 hours in 1995). The ratio between debriefing time and write–up is about 1:2 i.e. each hour of debriefing requires a further two hours of formulation, research and writing. The technical account based on one-to-one debriefing sessions is within an estimated 6,000 words of completion (2,000 words outstanding for the stereotype apparatus and 4,000 words for the final section which will describes the assembly of the engine). The writing rate is typically 180–200 words per hour.

The order in which the sections are presented is in the reverse order of writing i.e. the earlier sections are last, the most recent first. The genre of writing is highly internalist and allows little linguistic licence. The earlier sections (appearing last) tend to be costive, taut and formal. As the genre developed the style, though inevitably still constrained, tends nonetheless to be slightly more expansive.

Doron Swade
25 February 1995

# Printing

## General

Drawing References (main drawings in bold):
BAB [A] 147, 161, 162, 163, 164, 165, 166, **172**, **173**, 174, 175, 176

Integral to the concept of the Difference Engine is the automatic production of results by mechanical means. Babbage's intention was that the 'unerring certainty of mechanism', as Lardner put it, would eliminate the risk of human error in the process of calculation, recording results, transcription into tabular form, typesetting using loose type, and finally, proof reading — all processes ordinarily performed manually.

The printing apparatus appears on the left hand side of the vertical framing and is directly coupled mechanically to the calculating section (163). The whole mechanism is often referred to for convenience as the printing mechanism, or printer. However, though the apparatus does produce a printed record on a paper roll, the primary function of the whole mechanism is stereotyping i.e impressing numerical results on paper mache or soft metal. The stereotyped plates were intended to be used as moulds for making printing plates from which the tabular results would be duplicated.

The overall mechanism consists of two distinct sections: the stereotype table apparatus, and the printer and stereotyping apparatus. The stereotype table, shown as the lower section left in 163, is larger front-to-back than the calculating engine (164) and is supported by its own cast stand which arches over the two long members of the engine base. The table apparatus automatically positions the trays of paper mache, or strips of soft metal, under the print heads for each impression of a result and automatically repositions the trays at the end of each calculating cycle ready for the next impression. The layout of the results on the page is programmable by interchanging a selection of pattern wheels. A variety of column formats are available. Single column layouts are produced by impressing results on successive lines; multiple column formats are generated in a column-to-column sequence across the page or as two vertical columns wrapping round from bottom to top. Separation between lines is also programmable in fixed increments determined by the pattern wheel selected and results an be stereotyped with large and small typefaces.

Only the 30 least significant digits of the 31 digits result on the figure wheels are transferred to the printing and stereotyping apparatus: the most significant digit, represented by the uppermost figure wheel on the tabular column, is not transferred and is therefore not printed or stereotyped. This is digit is probably intended as a guard digit to act as an overflow warning.

The printer and stereotyping section, shown as the rectangular assembly above the table apparatus (163), contains sets of number wheels which are lowered once each cycle to impress the results on the soft material below. A full 30–digit result is impressed in one action i.e. all digit positions are impressed simultaneously. Babbage calls the number wheels stereotype sectors (172) as a portion of the circumference of each wheel has embedded in it ten heads embossed with the numerals 0–9 (172). The stereotype sectors have two degrees of freedom only: rotational and vertical i.e. each sector can rotate about a horizontal axis to position the correct head for a digit, and the whole assembly is lowered to impress the result. The stereotype sectors are not free to move in the horizontal plane i.e. the position of the result on the page is determined by the X–Y motion of the stereotype table. The apparatus provides options for two type sizes and spacing of print. The two print pitches are fixed i.e. 1/8" and 1/16". The two sets of print heads act in tandem i.e both are lowered together regardless of which is actively impressing the material in the trays. Impressions from both sets of stereotype sectors can be obtained in the same run.

In addition to the stereotyping function, the printer section prints a hard–copy record of each result as part of the stereotyping cycle. The tabular result is transferred to print wheels (printing sectors) at the same time as the stereotyping sectors are set. Theapparatus contains a paper roll feed, an ink bath and a system of inking rollers, and printing sectors which automatically produce an ink–on–paper printout, one result per line, on a continuous paper roll. The roll is automatically advanced each cycle. The printer is a valuable aid for testing and during setting up; it also provides a hardcopy record of all results.

The printing and stereotyping apparatus is driven by gears pinned to a shaft running the full length of the engine below the calculating section. The drive for the shaft is derived from the main cam drive shaft via a bevel gear on the underside of the cam stack (163). There is no printing overhead in the timing cycle i.e. the stereotyping and printing action does not lengthen the timing cycle but takes place in parallel with the second half of the second half–cycle.

## Operation

*General*

The tabular value is transferred from the last even figure wheel column to the printing and stereotyping apparatus by a transmission train consisting in turn of the tabular figure wheels, sector wheels, horizontal racks, pinions, and horizontal spindles (174, and 176 bottom left). The spindles drive a set of vertical racks via a second set of pinions. The vertical racks project downwards into the printing and stereotyping apparatus and form the final link in the transfer from the calculating section.

The general operation of the printing and stereotyping apparatus is shown on elevation 172 and plan view 173. The lower section of the vertical racks mesh with the circular printing sectors and the downward motion of the racks transfers the thirty–digit number to the set of printing sectors. The anticlockwise displacement from zero of the printing sectors positions the print heads in a straight line at three o'clock ready to receive the impression first of the inking roller, and subsequently of the paper roll. The trajectory of travel and impact position of the inking roller and paper roll is shown dotted in 172.

The printing sectors drive a set of horizontal racks (called stereotyping racks) which run front to back. Geared to the underside of the racks are two sets of stereotyping sectors. The right hand set have a fixed 1/8" pitch; the left hand set have a 1/16" pitch. The narrowing of the horizontal racks to achieve the reduced pitch is clearly shown in plan 173. The vertical racks drive the printing sectors, which drive the horizontal racks, which in turn advance the two sets of stereotyping sectors. The tabular result is transferred to both sets of thirty stereotyping sectors simultaneously. The motion of the racks positions the two rows of print heads facing downward at six o'clock. At the appropriate point in the cycle both sets of stereotype sectors are lowered to impress the soft material in the trays on the stereotyping table below. The stereotype sectors separate from the racks during this process and re–engage on completion.

Provision is made for locking various pieces of the apparatus. V–shaped locking notches and locking bars are shown operating on the vertical racking, and the printing sectors. Locking notches are also shown between the type heads on both sets of stereotyping sectors. A claw–lock mechanism is shown for the small stereotype sectors. Despite appearances only one locking bar is active at any time: depending on displacement of the stereotype sector either the left or right locking bar engages with the v–notches between the type heads. The left hand bar does not operate on the sector gear teeth. The detail of this locking mechanism is not redrawn for the large stereotype sectors, though it is clear from the provision of locking notches between the type heads, mountings for the lock mechanism (172), as well as the dovetail slider and forked drive arm (173), that the intention is to lock the large stereotype sectors in this way. In all, the printing and stereotyping mechanism is locked at five separate points in the train. The fifth lock is the vertical bar locking sector racks (174).

The apparatus for producing a hard–copy record on a print–roll is shown on the right of 172. The cluster of rollers (top right) is an inking system. The final inking roller is lowered onto the line of print heads after the result has been set up. The apparatus below is a paper feed system which incrementally advances the paper each calculating cycle and propels the paper roll in a circular trajectory onto the inked print head to provide a continuous printed record of results.

The circular apparatus shown left centre in 172 is a side elevation of the cam stack which provides the drive and phasing for the various operations. The main drive wheel is shown inside the frame at the top of 173. This drives the main cam shaft and the cam stack consisting of fourteen cams clustered at the top of 173 as well as one additional eccentric, outside the frame at the bottom of the drawing, which drives the links to propel the ink roller.

A plan view showing the layout is given in 173. The small stereotype sector assembly is shown on the left and the large sector assembly is shown centre right. Both sets of sectors have dovetail sliders which allow the vertical punching motion required for stereotyping. The top bar of the $\Pi$ – shape (***Q***) drawn over the printing sectors (***R***) is the locking bar which traverses the full run of aligned v–notches cut into the top of the printing sectors.

*Coupling to the Calculating Section*

The result on the tabular column is transferred to the vertical stack of 30 horizontal racks of the printing apparatus by an intermediate sector axis ($\boldsymbol{R}^0$, 176, 161). This last sector axis, interposed between the racks and the tabular column, is an even sector axis i.e. the reverse–handing of alternate axes leaves this last sector axis unchanged. The tabular value is

given off to the even sector axes, $\boldsymbol{R}^0$ included, during even to odd addition i.e. at the start of the second half cycle. Each sector of $\boldsymbol{R}^0$ engages with a horizontal rack (176, 161).
The arrangement translates the anticlockwise rotation of the sector into a linear displacement of the rack in which the travel of the rack is proportional to the tabular value.

The $\boldsymbol{R}^0$ sectors act as normal even sectors i.e. they restore the tabular value after giving off in the same way and at the same time as the other even sector axes i.e. in the half–raised position; the rotational travel of the drive arm for this last even sector is the same as for the other even sectors. The return of the sectors to the zero both restores the tabular value and returns the racks to their home positions.

Though the last even sector axis functions as a normal even axis, the $\boldsymbol{R}^0$ sectors differ from those of the other three even axes, $\boldsymbol{R}^2$, $\boldsymbol{R}^4$, and $\boldsymbol{R}^6$, in respect of depth of teeth, number of teeth and angular orientation. The opened out view at the top of 171 shows the standard calculating sectors with two runs of teeth of different depths: the run of teeth that engage with the figure wheels to the right are full depth, and those that engage with the figure wheels to the left are single depth so as to allow partial disengagement in the intermediate vertical position (see Calculating section). However, both runs of teeth on the $\boldsymbol{R}^0$ sectors are full depth. This is shown in elevation 174 top right. Here the sectors, shown fully raised, are disengaged from the tabular figure wheel but still engaged with the racks. The $\boldsymbol{R}^0$ sectors in fact never disengage from the racks. The $\boldsymbol{R}^0$ sectors have one more tooth than the standard sectors. This is clear from the 176 bottom left which shows 13 $\boldsymbol{R}^0$ sector teeth for engagement with the rack compared to 12 for sectors on other axes. Finally, the angular orientation of the rack sector teeth is different (176, 161): 161 shows the limits of travel of the $\boldsymbol{R}^0$ sectors bounded by the solid line at eight o'clock and the dotted line at 27 minutes past; the limits of travel of the other even sectors is shown as bounded by the solid line at 10 o'clock and the dotted line at 7 o'clock. The sector zero stop (${}^2\boldsymbol{M}^0$) is shown scalloped out to accommodate maximum displacement of the sector.

Each sector rack (called figure racks in the Timing Diagram 385a) have two sets of teeth at right angles: vertical teeth in the horizontal plane, and horizontal teeth in the horizontal plane. The horizontal teeth mesh with the $\boldsymbol{R}^0$ sectors; the vertical teeth mesh with a stack of staggered pinions. The pinions drive a set of horizontal spindles fitted with another set of pinions at the left hand ends. These mesh with a set of vertical racks shown in elevation in

174 left and in plan in 173. The overall function of the arrangement is to transfer the displacement of the tabular figure wheels into vertical displacement of the racks. The sector racks are shown in plan on 176 bottom left, and both sector racks and vertical racks are shown in elevation on 174. The pinions are staggered front-to-back and top-to-bottom i.e. they are arranged vertically in pairs with the even numbered digits (**N**) offset upwards and backwards (176, 174 and 172). The odd-numbered pinions engage with the racks below; the even-numbered pinions engage with the racks above (174). The odd- and even-numbered racks are different so as to accommodate the horizontal staggering and vertical pairing. (The racks are detailed in modern drawings K312, 313; horizontal staggering was dispensed with.)

The elevation in 174 shows the four lower-most tabular figure wheels (units, tens, hundreds and thousands) coupled digit-for-digit to the vertical racks. The upper section of the tabular column is not shown. Though the tabular column records a thirty-one digit result, there are only thirty vertical racks shown in elevation 174 and plan 173. Only thirty digits of the thirty-one-digit result are printed i.e. the result from the top-most tabular figure wheel in not transferred but probably used as an overflow check.

The v-shaped notches on the sector racks shown in 176 and 161 are locking notches for the vertical locking bar. The operation of the lock is shown in elevation in 174. The single locking bar serves all thirty digits. The lock is engaged by the downward motion of link arm $^{6}$***H***. This operates drive link $^{4}$**V** turning on a fixed pivot, which lowers and engages the lock. No detail of the upper pivot is shown. A slave rocker, which mimics the action of the right hand half of the drive link, was added in the upper section to guide the lock and ensure that it remains vertical during the locking and unlocking action.

*Vertical Racks*

Each vertical rack has two short runs of teeth, one set in the upper section, and one set in the lower section. The teeth in the upper section mesh with the pinions driven by the sector rack spindles (174) and the lower teeth mesh with the printing sectors (172).
The overall arrangement transfers the circular motion of the tabular figure wheels to circular motion of the printing sectors.

Though not shown it is assumed that all thirty vertical racks are run to the full height of the machine. The racks slide against each other and the whole set is sandwiched between two vertical framing side-pieces $^{4}$**P** and $^{5}$**P** (174) which act as constraining cheeks. (Framing pieces are not shown in 174.) The racks are further constrained by a guide–bar which threads through slots at the lower end of all the racks. The guide–bar is shown clearly in 174 fitted into two counter bored slots in the side pieces. The guide–bar both constrains the rack motion to the vertical and acts as a tie–rod for the assembly. 172 centre shows the guide slots in the racks and rectangular cross–section of the guide bar (**U**).

The nominal width of the rack pinions is 1/4" (manufacturing width is specified slightly under for clearance) and the pitch of the racks is 1/8" i.e. there are eight racks per inch. The increased width for the rack teeth is achieved by lapping adjacent racks in two mating L–shapes. Detail of the lapping appears in 173 which shows two pinions (digits one and two) and three lapped pairs (digits one through six) in cross section. The effect of lapping is to double the effective width of the gear teeth to twice the rack width. As a result of lapping the rack teeth and pinions for the thirty odd–numbered digits face the front of the engine, and those for the thirty even–numbered digits face the rear (173). Because of the mirrored arrangement of pairs of sector racks, alternate spindles rotate in opposite directions (174) i.e. the odd–numbered spindles rotate into the page and the even numbered spindles out of the page. The lapping arrangement corrects this so that the uniformly counter–clockwise displacement of the tabular figure wheels from zero results in a uniform downward displacement in any of the associated vertical racks.

In addition to lapping, alternate rack pairs are offset. The intention here is for pairs of racks to act as guide channels for the pinions in between so as to reduce the risk of pinions fouling adjacent racks due to side play in the spindles. The system of offsetting is repeated below where the racks act as guide channels for the printing sectors (173). Though both upper and lower sections of the rack are offset, only the upper sections, which mesh with the pinions, are lapped. This can be seen from the cross–section view in 173.

The vertical distribution of the pinions and spindles follows the vertical pitch of the figure wheels. A length of toothed section is shown in 174 which indicates that the upper sections of the racks are geared only where needed as shown for the lower rack sections in 172. The positions of the toothed sections on the upper portions of the racks will therefore be staggered diagonally bottom right to top left across the face of the rack assembly.
The vertical racks are therefore not identical. The pitch of the vertical lines in 174 is that of a single rack width (1/8"). Strict interpretation of 174 would indicate that the lapping is

confined to the toothed sections only since, if the lapping extended, alternate vertical lines would be dotted. However, the use of broken lines to indicate hidden edges is not consistent elsewhere and 174 was not taken as a conclusive indicator that lapping should not be extended as guides for the sliding motion (see Implementation). (K331 A–Q gives modern detail of odd digit racks; K332 A–Q gives evens).

The upper section of the rack assembly is not shown. The guide bar assembly which constrains the lower section of the racks was not duplicated here. Rather the upper section of the racks are trapped front–to–back between the two sets of pinions which act as a restraining cage. The intention to offset alternate racks front–to–back is confirmed in 172 centre top which shows the lowermost four spindles and pinions for the first two digit positions. The spindles of the upper pair (digits three and four) are shown offset to the right. This offset helps to clarify an apparent discrepancy in the front–to–back spacing of the spindles. The separation of spindles for the first odd and first even digits is consistent in 176 bottom left, 173 centre top, and 172 centre top. 161 bottom left shows this separation as slightly wider. However, the wider spacing in 161 is consistent with the spindle positions for the second and third digits rather than the first and second digits i.e. the positions shown in 172 for the spindles bottom left and top right. [Is M and N notation generic or decade dependent?]. (The ***M*** notation indicates even–powered digits (units, hundreds, ...);
the ***N*** notation indicates odd–powered digits (tens, thousands, ....)

*Implications of Rack Offset*

Since the printing sectors are concentric and pivot on the same shaft, offsetting the vertical racks alters the pitch circle radius of alternate sectors. Equal downward displacements of the vertical racks will therefore result in different angular displacements of alternate printing sectors. A 0.2" difference in pitch circle radius (the size of the offset) results in about a 2.5° difference between the angular displacement of a sector set to print a zero and one set to print nine. If the pitch of the type heads were identical for all the printing sectors then the nine would be positioned as much as a full character height above an adjacent zero. This difference is clearly too large to be ignored. The implication is that the pitches of the locking notches, and that of the type heads, need to be altered for alternate printing sectors to ensure that all the print heads line up on the print line and all the locking notches are aligned along the line of the common locking bar whatever numbers are registered on the sectors. Only one printing sector is shown in elevation (172) and there is no indication of any provision for non–identical sectors.

If offsetting is regarded as a desirable aid to maintain alignment between the printing sectors and vertical racks, then it is, by the same reasoning desirable for the meshing of the printing sectors with the horizontal racks. Offsetting would also be desirable for the meshing of the stereotyping sectors with the lower edge of the horizontal racks. Here both sets of stereotype sectors separate from the horizontal racks when lowered to make an impression, and it would seem that the guide channel effect of offsetting alternate racks would assist alignment during re–engagement. The implication of offsetting alternate horizontal racks is that the pitch of the type heads would need to be different to ensure that they line up at six o'clock for all the sectors whatever the number set up. As with the printing sectors, there is no evidence in 172, or elsewhere, of provision for non–identical stereotype sectors.

The issue of how far into the drive train to extend the rack offset is resolved by the geometry of the stereotype sectors. In the case of the printing sectors the run of locking notches is separate from run of type heads. However, in the case of the stereotype sectors the locking notches alternate with the type heads on the circumference of the sectors. Increasing the pitch of the type heads on alternate sectors to compensate for rack offset so that the type heads align at the six o'clock position for any combination of numbers conflicts with the alignment of the locking notches for the correct operation of the common locking bar operating at the twenty–past the hour point. The two separate conditions of a fixed print position and a fixed separate locking point cannot both be satisfied if the pitch of alternate sectors differs *and* the type heads alternate with the locking notches. In the case of the printing sectors the two conditions are reconciled by placing the sequence of type heads and locking notches into separate groups. In the case of the stereotype sectors there is insufficient space on the circumference of the significantly smaller stereotype sectors to accommodate separate runs of type heads and locking notches. The conclusion is that the lower sections of the horizontal racks and the stereotype sectors cannot be offset.

The upshot of this conclusion is that the offsetting alternate racks must be extended to the gearing between the printing sectors and the horizontal racks to correct for the differing angular displacements of alternate printing sectors resulting from offsetting alternate vertical racks. Offsetting the gearing between the printing sectors and the horizontal racks ensures that the horizontal displacement of all stereotyping racks is the same for equal displacements of the vertical racks. Since the pitch circle radius of the two runs of teeth of each of the printing sectors is the same for any single printing sector (though different for alternate sectors) carrying the offsetting through to this last stage results in a 1–1 relationship between vertical rack displacement and horizontal rack displacement for all thirty positions. Again there is no indication of any provision for non–identical racks or printing sectors in the original design.

*Framing and Horizontal Racks*

The overall structure of the printing and stereotyping sector assembly is that of a large rectangular box frame made up of cast framing members bolted together. The frame is formed at the narrow ends by two end pieces $\mathbf{N}_1$ and $\mathbf{N}_2$, and by long side pieces, $\mathbf{M}_1$ and $\mathbf{M}_2$ (173).

The horizontal racks are boxed in at the full width end by the main side piece framing members ($\mathbf{M}_1$, $\mathbf{M}_2$). At the narrower end the straight sections are sandwiched between two edge blocks. The curved sections are contained sets of keepers, four above (${}^5\mathbf{Q}_2$, ${}^6\mathbf{Q}_2$, ${}^7\mathbf{Q}_2$, ${}^8\mathbf{Q}_2$) and four below (${}^1\mathbf{Q}_1$, ${}^2\mathbf{Q}_1$, ${}^3\mathbf{Q}_1$, ${}^4\mathbf{Q}_1$) (172), which span the main framing members, $\mathbf{M}_1$, $\mathbf{M}_2$. The straight sections of the horizontal racks are in contact and slide against each other i.e. they rely on each other for correct registration. The curved sections have small tapered clearances between them and will touch during operation depending on the combination of numbers in the result. The curved sections of the racks are formed in the shapes shown and retain their curvature when relaxed and do not rely on lateral contact to retain their form i.e. the curvature is not a sprung form resulting from blocking the straight ends of flexible strips.

**Paper Printer**

*Inking*

The provision for inking the printing sectors is shown on 172 top right. The general principle of operation is shown in a diagrammatic way as a cluster of rollers but the drawing gives no detail of drive or support. In this respect, the inking mechanism is less detailed that the rest of the printing and stereotyping mechanism. A small amount of additional detail is given in 173 which shows the inking roller supported between two levers (${}^1\boldsymbol{K}_1$, ${}^1\boldsymbol{K}_2$).

The elevation in 172 shows four rollers. The ink pot roller ($\boldsymbol{Z}$) has a scraper plate bearing on the axial surface of the roller. The scraper, roller, and side cheeks (latter not shown), form a reservoir which holds the ink supply. Printing ink is typically viscous and the arrangement, standard at the time, is sufficient to prevent leakage of ink past the scraper. The ink bath is shown fitted with a hinged dust cover.

The ink–pot roller bears on a second roller (***J***) which in turn bears on a third roller (***K***) which effects the final transfer of ink to the inking roller (***I***). Three of the rollers are shown suspended in space with no visible means of support, and there is no detail as to the method of drive, whether intermittent or continuous, or any indication as to the composition of the roller material.

The final inking roller is propelled onto the line of type set up with the thirty–digit result. The inking roller swings on two arms $^{1}\boldsymbol{K}_{1}$, $^{1}\boldsymbol{K}_{2}$, pinned to a shaft ($^{1}\boldsymbol{K}$) which is supported at each end by the two outer framing members ($^{1}\mathbf{J}_{2}$, $^{2}\mathbf{J}_{2}$) i.e. the shaft spans the full width of the whole apparatus. The shaft is driven by a drive arm ($^{1}\mathbf{K}_{2}$) (173 right bottom) which is in turn driven by link ***P*** which pulls the roller onto the line of type. The arc of the roller trajectory is shown in 172.

The provision of no more than a skeleton outline of the system of inking rollers (172 top right) could arise from the assumption of well–known contemporary practice. Given the degree of detail elsewhere, it is more likely that the omission of detail is an incompleteness in a partially finished drawing. The missing detail was supplied by reference to contemporary practice (see Implementation).

*Printing Stroke*

The basic printing and paper feed system consists of two drums or barrels, and a print roller. The general arrangement of the mechanism is shown in elevation on 172 with a partial plan in 173. A more detailed plan section of the take–up barrel assembly is given in 165: 'Elevation', Fig. 7, and end view, Fig. 6. The 'Elevation' shows the printing mechanism in fully raised position as viewed from the front of the engine i.e. looking left to right at the 'End View' shown in Fig. 6.

The paper path is shown in 172 and 165, Fig. 6. Paper is threaded from the roll wound on drum $^{1}\boldsymbol{F}$, around the print roller, ***F***, and onto the take–up drum $^{4}\boldsymbol{H}$. The rest position of the assembly is as shown in 172 and in the dotted position of 165, Fig. 6. The relationship between the axes of the two drums and the roller is fixed i.e. the two arms, $^7\mathbf{D}_1$ and $^7\mathbf{D}_2$, carry the shafts of the drums and roller, determine the relative positions. To print a result the whole mechanism turns on the fixed pivot ($\mathbf{D}^7$) and propels the printing roller in a circular trajectory to press against the line of type heads. The position of the assembly at the completion of the upstroke is shown as the solid view in 165, Fig. 6. During the printing stroke the whole assembly moves as a unit i.e. there is no relative motion between the drums and the roller.

The upstroke is driven from the single heart–shaped cam, $^2\mathbf{A}_4$, shown in elevation on 175, and in plan as part of the cam stack towards the top of 173. A train of links and arms transmits the drive to the drum assembly. The angle of the elbow of the follower arm (***H***) is fixed and the two fixed pivots in the chain are the follower arm pivot ($^4\mathbf{B}$) and the take–up spool axis, $^7\mathbf{D}$ (172, 173). The two floating pivots are the forked knuckles at the end of drive link $^2\mathbf{J}$. The follower arm, **H**, pivots on $^4\mathbf{B}$ and drives link ***J*** diagonally downwards, and arm $^7\mathbf{C}$ (173), keyed to shaft $^7\mathbf{D}$, levers the drum into the upstroke.

The drive is unidirectional i.e. the return stroke is not actively driven. The assembly passes top dead centre during the upstroke (165, Fig. 6). To avoid the assembly being stranded in the printing position a counter weight is provided initiate the return stroke. The counter–weight, $^7\mathbf{C}^2$, is shown as a circle on 172 right bottom and is fixed to a lever arm attached to the boss of $^7\mathbf{C}$ (173). The counter–weight acts on the shaft on which the whole mechanism pivots and the return stroke is a controlled descent restrained by the return pressure of the cam follower on the retreating profile of the heart–shaped cam (175). During the return stroke the print roller retraces the trajectory of the upstroke. The rest position at the end of the return stroke is determined by the end stop $^7\mathbf{C}^{3?}$ which bears on the vertical frame (172). The plan view (173) does not show the end stop lever but the notation indicates that it is part of the ball–weight lever assembly ($^7\mathbf{C}^2$).

*Paper Feed*

Description of the paper feed requires further exploration of the assembly shown in section in 165, Fig. 7. Each of the two swinging arms, $^7\mathbf{D}_1$, $^7\mathbf{D}_2$, which support the drums and roller, are integral with a long boss. Each boss is pinned to the shaft $^7\mathbf{D}$, outside the swinging arms. The bosses extend inside the arms, sleeving the shaft, and meet in the middle of the assembly. The boss on the side with the ratchet gear has a cam fixed to it (165, Fig. 7 left), the outline of which is shown in two positions (dotted and solid) in 165, Fig. 6. A second assembly is free to rotate on the shaft formed by the boss. This consists of a sleeve, the ratchet gear, and the gear wheel (Fig. 7 right). The gear wheel meshes with a similar gear wheel fixed to the feed drum. The third part of the nested assembly is the barrel of the take–up spool. This rotates on the sleeve which carries the ratchet and gear wheel. Trapped in a recessed annulus between the take up spool and the sleeve is a clock spring (Fig. 7, right) pinned at one end to the inside of the take–up barrel and the sleeve at the other. The wound clock spring biasses the take–up spool to rotate anticlockwise (172) and maintain positive tension on the paper. A feed pawl ($^{5}$***L***) pivots on the question–mark bracket $^{3}$***K*** (172) and is biassed by a leaf spring ($^{5}$***E***) to engage with the ratchet teeth. The feed pawl is shown engaged in 172 and lifted clear by the cam in 165. The final piece of the assembly is a backstop pawl (172) which engages with the ratchet teeth under pressure from a second leaf spring. There is no provision in evidence for fixing the paper roll at either end. In the rest position the feed pawl engages with the ratchet teeth (172, 165 Fig 6). During the printing stroke the feed pawl is lifted clear of the teeth as shown by the solid view in 165, Fig. 6. The backstop pawl holds the ratchet and gear assembly in fixed relation to the swinging arms during the upstroke. The backstop pawl ensures that the assembly moves as a unit during the printing stroke with no relative motion between the drums.

Towards the end of the return stroke the cam releases the feed pawl which engages the ratchet (dotted view 165 Fig. 6). This halts and holds the ratchet and gear wheel in the position at engagement. In returning to the rest position the remaining travel of the swinging arms drives the gear of the feed drum against the stationary gear wheel of the take–up assembly i.e. the feed drum assembly and stationary gear act as planet and sun gears. The ratchet overrides the backstop pawl which remains in sprung contact throughout. The feed drum is driven clockwise and feeds a short length of paper from the roll. The paper is taken up by the take–up spool driven by the clock spring and tension is maintained. The overall action presents a fresh area of paper on the printing roller ready for the next impression. The length of paper incrementally advanced depends on the fixed angle through which the feed drum is driven at the end of the return stroke, and the diameter of the paper roll on the feed drum. As the paper stock depletes, the length of paper fed each printing cycle, and therefore the line height of the print, will vary slightly. The direction of the variation will be to reduce line height as the calculation run progresses.

*Paper Feed – Design*

The design of the paper feed arrangement is of interest. The unloaded diameter of the feed and take–up drums is shown in 165 Fig. 7 as 3" (The note written in the rectangle representing the feed drum confirms that the barrel as drawn is unloaded: "Printing paper coiled on this barrel at the commencement".) The two views (165 and 172) showing the routing of the paper between the drum also show the diameters as 3" with no indication of the effect on the diameters of paper stock. The 174 plan view shows what appears to be a single thickness of paper stretched between the roller and the take–up spool. This suggests that the printout is intended for short print runs only (checking and testing) and that the printer roll would therefore be replenished each time a stereotyping tray is changed. If only a few turns of paper are loaded at time then the effects of the gradual change in the effective diameters as the reserve stock transfers from the feed roll to the take–up drum can be ignored.

The housing for the clock spring (165, Fig. 7) is relatively small (the height of the channel is about 0.5") and the modest number of turns of the take–up drum available to pre–tension the spring would seem to limit the length of paper the spool can take–up. A further concern is that the tension of the clock spring may override the backstop pawl and cause forward runaway. The feed pawl is little use here and is in any event lifted clear of the teeth during the upstroke and the backstop pawl offers the only resistance to runaway during this part of the cycle.

The maximum capacity of the printer is worth further exploration as it exposed a subtlety in the design that was not at first obvious. The maximum paper stock possible is physically limited by the gap between the feed drum and take–up spool. This gap is about 0.58" (165, Fig. 7). With paper thickness of 0.004" the number of turns is approximately 145 and the approximate length of the reserve paper roll is at least 113 feet (assuming a constant nominal barrel diameter of 3" and zero stacking factor). This upper limit is unnecessarily long and can happily be reduced by a factor of four with the added advantage of making the loading the paper stock (rolling on manually) more manageable. A radial thickness of 1/8" of paper provides an unbroken run of about 24 feet which is a practical length and will serve to pursue the ball–park design calculation.

A 24–foot run of paper corresponds to a nominal 31 turns on the feed drum. The size of the annular channel for the clock spring is seemingly too small to house a spring of sufficient length to allow 31 turns for pre–tensioning. However, closer examination of the operation of the take–up mechanism shows that the total number of turns of the take–up spool to transfer all the paper stock from the feed roll is only a small fraction of the number of turns of paper on the roll at the start. When the feed pawl engages during the latter part of the return stroke the feed drum rotates clockwise against the stationary take–up gear so as to issue a short length of paper. The downward arc of the printing roller wraps a length of paper around the take–up drum.

At the start of the run the diameter of the feed roll will be greater than that of the take–up barrel; when the paper stock distribution is equal the diameters will be equal; and at the end of the run the diameter of the take–up drum will be greater. If the take–up drum was not free to move then the incremental length of paper issued each cycle at the start of the run will be slightly longer than that wrapped around the take–up drum during the return stroke. Similarly, if the take–up drum was not free to move then the incremental length of paper issued at the end of a run will be slightly less than that wrapped around the take–up drum. The take–up drum therefore only takes up the small differences that result from the incrementally changing diameters as the paper stock transfers between drums. At the start of the cycle, the compensatory motion from the clock–spring will be clockwise; when the paper stock is equally distributed between the drums there will be no compensatory motion; at the end of the paper supply, the compensatory take–up motion will be anticlockwise.

The worst case difference occurs at the start and end of the paper supply when the distribution is most unequal. For the purpose of estimating the number of turns required of the take–up spool assume that this worst case situation persisted throughout the paper transfer. The difference between the length of paper issued and wrapped for a paper stock 1/8" thick in one turn of the feed drum will be $\Pi$ x(difference in diameters) = $\Pi$ /4 = 0.785" per turn. Since the direction of compensation reverses at the half way point of the stock transfer, the number of take–up turns required is half the total number of turns i.e. 31/2, say 15 turns. The worst case length discrepancy over the whole 24–foot roll will be 15x0.785"=12" (say). The number of compensatory turns by the take–up spool in either direction will therefore be 12"/( $\Pi$ x3")=1.27 turns. This is well within the capacity of the clock–spring. This estimate assumes worst case diameter difference of 1/4" for the whole transfer. The reality is much more favourable than this. The upshot of this analysis is that the clock–spring is not the limiting factor in the take–up capacity of the mechanism. Due to the wrapping action of the downstroke the relationship between feed turns and take–up turns is not one–to–one as first appeared.

For a 1/8" thickness of paper stock the line feed with a fresh roll is $\Pi$ D/54 = 0.189" where 54 is the number of ratchet teeth. The worst–case difference in line–height in the printed result (the difference in line height between the start and end of the 24–foot roll) due to the shrinking diameter of the feed drum as the paper stock depletes is (2 $\Pi$ /54)x(thickness of paper stock) = 0.0145". Given that the ratchet advances one tooth per cycle this corresponds to about 8% of the nominal line feed.

*Locks*

Locks are provided at five separate points in the transmission train: the figure racks (176, 174), the lower sections of the vertical racks (172, 173), the printing sectors (172, 173), and both sets of stereotyping sectors (172, 173). In all instances locking is by means of a single common locking bar engaging with an aligned set of v–shaped notches. The locking bars for the sector racks, vertical racks and printing sectors are operated by pivoted levers driven directly by cam followers.

The horizontal racks driven by the figure wheel sectors are shown in plan in 176 (bottom left) and in elevation in 174 (top right). These views show a single wedge–shaped lock serving the vertical stack of thirty racks. The lock is lifted and withdrawn by the angle crank turning on fixed pivot ${}^{1}\boldsymbol{V}$. The crank is driven by link ${}^{6}\boldsymbol{H}$ (174) which is in turn driven by the cam follower rocker arm from conjugate cams ${}^{2}\boldsymbol{A}_{2}$, ${}^{2}\boldsymbol{A}_{3}$. The axis of the pivots at each end of link ${}^{6}\boldsymbol{H}$ are at right angles to each other. This practice is questionable but the assumption is that the deviation from linear drive is sufficiently small to be accommodated by play in the pivot. 174 does not show the upper section of the lock and therefore details of how the lock is constrained to the vertical during operation are not given. To guide the lock the twin–armed lock lever (${}^{4}\boldsymbol{V}_{1}$, ${}^{4}\boldsymbol{V}_{2}$) and mounting bracket shown in plan in 176 were duplicated in the upper section but the drive link, ${}^{6}\boldsymbol{H}$, was not extended. The motion of the lock is therefore fixed by the radial locus of the upper and lower lever arms only the lower one of which is driven.

The locking mechanism for the stereotyping sectors requires additional explanation. 172 shows what appears to be a claw–lock mechanism for the small stereotype sectors (172 bottom left). However, despite appearances only one locking bar is active during locking. The bars close simultaneously and depending on the position of the type wheel, one or the other of the bars will engage with the v–notches between the type heads. In the position shown in 172, the left hand locking bar does not engage with the sector gear teeth. The lock is operated by raising the twin–tyned locking slider. The angled shoulders of the slider drive the two upper curved followers outwards and bars close in a rocking action. Lowering the slider operates the lower two curved followers to release the lock. The locking slider rides on a single dovetail slide ($^{1}\boldsymbol{y}_2$, 173) and is driven by forked lever $^{3}\boldsymbol{X}$ pinned to live rocker shaft $^{3}\boldsymbol{G}$. The rocker shaft is driven by conjugate cam pair $^{2}\boldsymbol{A}_{11}$, $^{2}\boldsymbol{A}_{12}$ (173) and contact followers $^{3}\boldsymbol{W}_1$ and $^{3}\boldsymbol{W}_2$ (172, 173).

The claw–lock is not redrawn for the large stereotype sectors though it is clear from the provision of the dovetail slider, cams and followers that the method of locking the large stereotype sectors is repeat of their smaller counterparts. The cams in this case are $^{2}\boldsymbol{A}_{13}$ and $^{2}\boldsymbol{A}_{14}$ and the followers, ?**W**.

*Cams*

173 shows the cam stack in plan and partial elevations and cam profiles are given in 172 and 175. The cam shaft drive wheel ($^{2}\boldsymbol{A}_7$) is driven from the main drive shaft via an idler (163). The cam stack consists of fourteen cams some of which are conjugate pairs. (This excludes the additional eccentric to provide drive for the inking rollers.)

*Figure Rack Lock*

The wedge–shaped lock is lifted and withdrawn by the angle crank turning on fixed pivot $^{1}\boldsymbol{V}$. The crank is driven by link $^{6}\boldsymbol{H}$ (174) which is in turn driven by the cam follower rocker arm from conjugate cams $^{2}\boldsymbol{A}_2$, $^{2}\boldsymbol{A}_3$. The cam profiles and follower arms are shown in 172 as part of the general arrangement and abstracted more clearly in 175.

*Paper Drum Lift*

The linkage driving the printing assembly is shown in elevation in 172 and is described above. The heart–shaped cam ($^{2}\boldsymbol{A}_4$) omitted in 172, and fixed–angle follower (***H***) is shown clearly in 175 right. The v–shaped roller follower pivots on rocker shaft $^4\mathbf{B}$. The cam pushes the roller out to the right during the lifting stroke. The return stroke is not actively driven but the printing mechanism is prevented from being stranded past top dead centre by the action of the spherical counter–weight, $^7\mathbf{C}^2$ (bottom right 172), which drives the return stroke which is controlled by the pressure of the follower on the retreating profile of the cam.

*Printing Sector Lock*

The printing sector lock cam ($^2\mathbf{A}_5$) is the T–bar cam shown at about 3–o'clock on 175 right with the follower arm and wedge–shaped locking bar top right of the same view. The follower and locking bar are duplicated in 172 but without the T–bar cam. The T–bar strikes the follower arm $^8\mathbf{P}$ and drives the live rocker shaft $^8\mathbf{L}$ (173) to which it is pinned. The locking bar assembly $^8\mathbf{Q}$, also keyed to the rocker shaft, is driven so as to drive the locking bar into the run of notches to immobilise the printing sectors. The drive is unidirectional and the assumption is that the lock is disengaged by the counterweight of the mechanism to the left of the rocker shaft after the T–bar has passed. Since the T–bar is only a partial cam the rest position of the follower arm is determined by which side of the pivoted mechanism is heavier. If the follower–arm assembly is heaver then the rest position will be the curved section of the follower arm resting on the rocker shaft for the stereotype sectors follower. The likelihood, however is that the locking bar side is heavier and the lock will bounce on the sectors as they rotate.

*Vertical Rack Locks*

A T–bar cam ($^2\mathbf{A}_6$) is shown at 6–o'clock in 175 left. The follower arm and locking bar are shown in elevation in 172, 175, and in plan on 173. (The bar is obscured in the plan view and is shown as a dotted outline.) The T–bar acts on the contact follower to engage the lock. The lock disengages by dropping out or being kicked out when the rack next moves.

*Stereotype Sectors*

The small stereotype sectors are raised and lowered by conjugate cam pair $^2\mathbf{A}_7$, $^2\mathbf{A}_8$ (173) via cam followers $^8\mathbf{D}_1$, $^8\mathbf{D}_2$ shown in elevation in 172. Bosses for the two forked end levers ($^8D_3$, $^8E$) are pinned to the live rocker shaft. The forked end levers raise and lower the stereotype assembly which runs in dovetail slides $^5U$, $^5V$ (173).

The arrangement is duplicated for raising and lowering the large stereotype sectors. The conjugate cam pair in this case is $^2A_9$ and $^2A_{10}$. Profiles and followers are shown in 175 and 172 and the forked end levers drive the assembly in dovetail guides as for the small stereotype sectors.

*Stereotype Sector Locks* (see also page 68 for general comments)

The twin–tyned locking slider (172) rides on a single dovetail slide ($^1y_2$, 173) and is driven by forked lever $^3X$ pinned to live rocker shaft $^3G$. The rocker shaft is driven by conjugate cam pair $^2A_{11}$, $^2A_{12}$ (173) and contact followers $^3W_1$ and $^3W_2$ (172, 173).
The claw–lock is not redrawn for the large stereotype sectors though it is clear from the provision of the dovetail slider, cams and followers that the method of locking the large stereotype sectors is repeat of their smaller counterparts. The cams in this case are $^2A_{13}$ and $^2A_{14}$ and the followers, ?$W$.

*Ink Roller Cam*

The ink roller cam ($^2A_{16}$), follower and drive links are shown in plan in 173 bottom centre and in elevation in 172. The cam arm sweeps against the scythe–shaped contact follower which rocks on the fixed pivot working against the action of the compression spring, $^1F$. A long link operates rocker shaft $^1K$ which operates the two ink roller arms $^1K_2$ and $^1K_1$. The travel of the inking roller towards the print heads is gradual and the restoration to home position is a snap action as the follower leaves the cam and the compression spring is released. The spring has two functions: it provides contact pressure between the inking roller and the sliding roller while in the home position; it restores the inking roller after the downward sweep to the type heads.

## Implementation

### *Inking*

Comparison with printing machinery of the time indicates that at least one roller in the inking train was imparted with reciprocal axial motion to ensure uniform spread of ink. The combined rotational and sliding motion was intended to eliminate dead spots which would remain as un–inked bands on the circumference of the rollers. A further standard practice was to alternate the consistency of the rollers: the finaking roller (***I***) would be soft to ensure uniform inking of the print heads, the ink pot roller (***Z***) would be hard so as to aid sealing against the scraper by providing a true surface, and intermediate rollers would alternate in consistency — hard–soft–hard etc. Hard rollers were typically made of steel, and soft rollers from a toffee–like composition.

The method adopted for imparting sliding motion to the roller was to fix two cams at each end of the roller shaft (Assembly Drawing K25). The cams are circular 360° cams with sections of left– and right–handed helices (detail K42) machined as one piece with the roller. The cams ride against two pegs fixed to a support bar above the roller. The angular position of the cams and position of pegs is such that the sliding roller is alternately driven left by one cam and right by the other as a consequence of the roller turning. The roller is driven by a separately by gears i.e. does not rely on rolling friction to rotate.

The possibility of eliminating the two intermediate rollers (***K, J***) and imparting sliding motion to the final inking roller (***I***) directly, was excluded: the plan view (173) makes no provision for sliding motion of the inking roller; there is in any event no room for the helical cams between the swinging levers supporting the inking roller. There is therefore the need for a separate sliding roller. This in turn entails the need for an additional transfer roller to maintain the practice of alternating soft and hard rollers. This reasoning provides a credible interpretation of the original roller arrangement shown in 172: the two intermediate rollers were therefore retained in the interests of both authenticity and practicality.

Since the final inking roller is soft, roller ***K*** should be hard and ***I*** [J?] soft. Since the soft roller has a less determined surface, is subject to greater wear, and in addition needs some pressure to ensure even surface contact on the two rollers it couples, the soft roller should have a floating bearing. Since the helical cam method of imparting sliding motion relies on a fixed relationship between the roller and the axis of rotation, ***K*** was selected as the sliding roller (steel), and ***J*** as the transfer roller (composition).

The assembly of the inking system is shown on K25. The cluster of rollers is supported by two cast mountings added to the castings for the printer framing members (detail K416 A&B). The detail of the mountings is copied from the mountings shown for the paper feed take up roller immediately below. The transfer roller (***J***) rests in a cradle which is part of a rocker mounting (detail K436). The roller has a counter–weight (K439) which allows the roller to bear with constant pressure on rollers ***K*** and ***Z***. Raising the counter–weight lowers the transfer roller clear of the other rollers and the cradle arrangement allows the roller to be lifted out for cleaning. The ink blade or scraper (K419) is mounted on bronze side cheeks (K418 A&B) which provide the side–walls of the ink reservoir and also seal against the curve of the roller. The side cheeks insert into slots in the mounting and the ink feed is adjusted by adjusting two jacking screws (K423).

*Drive*

A review of contemporary inking arrangements indicated that inking would be too heavy if the ink feed was continuous. Intermittent drive for the ink pot roller was derived from an eccentric added to the main printer cam shaft (***D***, 173) outside the inking roller drive link (bottom of 173). The arrangement is shown in assembly drawing K25 top left). The reciprocating motion of the link operates a pawl which advances a ratchet wheel (K447) mounted on the roller assembly mounting plate K416A. The amount of rotation of the ink pot roller during each printing cycle is adjustable by altering the position of the sliding pivot on the slotted lever (K446) using the knurled thumbscrew. This alters the pull–back of the pawl and the motion transmitted to the roller. The ink pot and sliding roller are actively driven from the ratchet wheel via a short shaft and gear wheels. Reverse rotation is prevented by a backstop pawl, K448. The other rollers are slaved i.e. do not have independent drives.

*Pinion Offset*

The plan view 173 clearly shows alternate racks and pinions in the upper section of the rack assembly to be offset. The supposed purpose of recessing alternate pinions between a pair of racks is to trap alternate pinions in the guide channels formed by the racks on either side so as to prevent them fouling adjacent racks in the event of drift in the spindles. The length of the pinion bushes, shown pinned to the spindles in two positions in 173, are about 2.5". Only the first two of these are shown and these are outside the stack assembly. With the arrangement as drawn there is barely any clearance between the bushes of the recessed pinions and the adjacent pair of rack that form the cheeks of the guide. This is further illustrated in 172 centre to which shows virtually no clearance between the cross–section of the spindle and the teeth of the adjacent rack. To provide adequate clearance would reduce the wall of the bush to 1/16". This was regarded as inadequate.

The solution adopted was to abandon offsetting alternate pinions. The double thickness of the teeth (1/4"), created by lapping, allows two measures sufficient to avoid risk of fouling: chamfering the pinion teeth, and slightly reducing the pinion width to provide clearance from the sidewalls formed by the racks. Eliminating the pinion offset removes the need to scallop out the vertical framing piece to accommodate the first few of the lower most pinion bushes until the stagger takes the bush clear of the framing member. The need for scalloping the frame member is as shown for the first pinion in 172.

The lower rack sections which mesh with the printing sectors do not have the benefit of lapping and the rack teeth only 1/8" across. Here the offset feature was retained to provide sidewall guidance for the printing sectors. The benefit of offsetting is to reduce criticality of manufacture and fitting by providing guide channels. In the case of the pinions, sufficient margins of clearance are available to justify dispensing with this feature to resolve the problem of wall thickness of the pinion bushes; in the case of the printing sectors, the offset feature was retained.

*Printing – Design*

Fig. 7 in 165 labelled 'Elevation' is the view of the printing assembly in the fully raised position, shown solid in the companion 'End View' (Fig. 6) alongside i.e. when the paper support roller is pressed onto the line of type. In Fig. 7 the take–up barrel is shown below, the feed barrel and paper support roller above and the feed pawl shown separately bottom left. The take–up barrel is the more complex assembly and consists of a series of sleeves and drums rotating around each other. Two sleeves, integral with the left– and right–hand swinging arms, are pinned to the shaft ($^7D$). The sleeves pass over the shaft and are shown meeting in the middle. The outer surface of the two sleeve halves provide a bearing surface for the next sleeve which has the ratchet and cam fixed at the left hand end and the drive gear at the right. Bearing on two shoulders of this ratchet sleeve is a thin–walled outer drum. The drive between the ratchet sleeve and the outer drum is via a clock spring housed in the annular chamber bounded by an end–plate disc and the chamber wall, *[4]H*. The clock spring provides a take–up tension for the outer drum.

Several details are omitted. No method of fixing the cam, ratchet or gear to the single sleeve is shown and therefore the intended order of assembly of the nested sleeves, and the sequence of operations by which the ends of the clock spring are to be secured, if originally considered, are not known. No provision is made for loading the paper stock onto the feed barrel. The feed barrel is not detachable ((172, 173) so loading using prepared stock on hollow formers was clearly not intended. The paper would need to be wound onto the feed drum from an external magazine. However, no means of turning the feed barrel is shown, and no method of fixing the ends of the paper roll to the feed barrel or to the take–up spool is indicated. The feed barrel would be wound anticlockwise during loading. Since the feed barrel is geared to the take–up spool, the feed pawl and back–stop pawl would need to be released during loading and there is no provision made for this.

Mechanisms for releasing the backstop pawl, loading paper stock from a magazine, and securing the two ends, were therefore added to the original design. The general assembly is shown in K24 and the with details given in K481A.

The preferred method of securing the paper is by a cross–clamp set as a tenon into an open slot running across the barrel. The paper is trapped under the clamp which is fixed at each end by two screws let into cast lugs in the barrel. The curvature of the outer surface of the clamp follows the barrel radius so that the cylinder of the spool, interrupted by the mortise slot, is restored with the clamp installed. The mounting lugs are mirrored on the opposite side of the barrel for balance.

Accommodating the cross–clamp entailed minor changes in some axial and radial dimensions of the assembly as well as modification to the assembly of the clock–spring chamber. Convenient machining the mortise for the cross–clamp requires runout off the end of the barrel. With the clock spring chamber integral with the barrel, the cross–clamp would form a segment of the chamber outer wall. This would expose the spring when paper stock is replenished and run the eventual risk of fouling the spring through the accumulation of paper dust and other debris. Instead of keeping the chamber and barrel as one as shown in 165, it was thought preferable to assemble the clock spring, chamber and gear as a separate unit. Separating the spring chamber simplifies manufacture of the main chamber of the barrel, and has the advantage of allowing runout when machining the mortise slot. Stopping the clamp short of the spring chamber prevents exposure of the spring that would otherwise occur when the clamp is removed for paper loading. The arrangement has the additional advantage of allowing the spring, gear and spring housing to be preassembled; the spring is rivetted at one end to the outer wall, and slotted at the other end into the inner sleeve. A further incidental benefit of separating the spring chamber is that with the cross clamp removed, the side of the spring housing acts as a location reference for positioning the paper during clamping.

The clock spring chamber is driven by a pin let into the barrel and protruding into the spring chamber wall. The larger diameter of the pin is secured in the body of the barrel with by an interference fit; the reduced diameter is a clearance fit into the chamber wall. The shoulder formed by the reduced diameter prevents the pin drifting into the spring housing.

With a type pitch of 1/8", 30 printed digits and 1/4" margin each side, the width of paper roll is 41/4". To allow clamping across the full 41/4" paper width and at the same time retain the original spacing of the two swinging arms, the clock spring chamber was crowded closer to the gear than as shown in 165 so as to clear the right hand edge of the clamp. The width of the clock spring and the size of the chamber remains unchanged.

The assembly of the ratchet sleeve is not clear from 165. The ratchet is shown as one with the sleeve; the gear is shown separate but no means of fixing to the sleeve is shown. The ratchet teeth need to be hardened. For ease and economy of manufacture the ratchet is made as a separate item from high grade hardenable steel and secured to the standard steel barrel with three screws and the gear is keyed to the ratchet sleeve. The ratchet wheel is therefore replaceable as a separate item. At the ratchet end, the steel sleeve runs on the cast iron boss of the swinging arm which is an approved match of materials.

To allow room within the original diameter of the outer barrel for the key and for the fixing lugs which secure the clamp, the arrangement of nested bearings was modified. The original design shows the outer barrel bearing on the ratchet sleeve and the full length of the ratchet sleeve bearing on the swinging arm sleeves which are pinned to the shaft and meet in the middle. The altered arrangement is shown in K24. Since there is no absolute need for the swinging arm sleeves to sheath the full length of the shaft, these two half–sleeves were dispensed with and reduced to a boss at each end, integral with the side arms, and pinned to the shaft. In the original arrangement the inside of the ratchet sleeve bears on the swinging arm sleeve along its full length. In the modified arrangement the swinging arm sleeves are shortened to bosses and the ratchet sleeve is supported by bearing surfaces at each end only: the boss is extended through the swinging arm at the ratchet end to provide one bearing surface; at the gear end the boss does not extend into the barrel chamber the second bearing is provided by a phosphor bronze bush inserted as a press fit to avoid a steel–to–steel bearing surface. Eliminating the two full half–sleeves allows the diameter of the bearing boss at the ratchet end to be reduced from 1 3/4" to 1 3/8". The extra internal space gained by eliminating the sleeves allows the cross clamp securing lugs and a keyway for the gear to be accommodated within the original outer dimensions of the barrel assembly.

The take–up barrel and the clock spring chamber are assembled as a sandwich between the two swinging arms which act as side pieces. The gear forms the outer radial wall of the spring chamber instead of the end plate (165) which was dispensed with.

The cross–clamp arrangement for securing the paper to the take–up spool is duplicated on the feed spool but without the complication of the clock–spring chamber. In this case the face of the gear boss serves as a location reference when loading the paper roll.

*Loading Paper*

In a relatively rare explanatory annotation we find the following inscription on the drawing for the feed drum: "printing paper coiled on this barrel at the commencement" (165, Fig. 7). Babbage is not forthcoming about how this is to be accomplished and special provision for loading was made. The arrangements for loading the feed spool from a magazine are shown in general assembly K24. Paper stock is supplied from a detachable magazine (K58). This consists of two side cheeks, separated by three spacers, a loose top roller and two wooden cradles which support the paper roll. It is not clear from 173 or 165 whether the printing roller is intended to rotate or not. The preference was to provide a fixed bar with a flat so as to act as an anvil for the type heads. The magazine is lowered onto the printing mechanism and locates in two additional slots machined into the paper support bar which is not free to turn.

The feed spool is wound anticlockwise during loading from the magazine. Since the take–up spool is geared to the feed spool, both the backstop pawl and the feed pawl need to be disengaged to allow free rotation of the barrels. The feed pawl is disabled by halting the calculating cycle before the printing mechanism is fully lowered i.e. before the feed pawl cam releases the pawl for engagement. No additional mechanism is required for this. There is no such advantage to be had when it comes to the backstop pawl which remains engaged throughout the normal calculating cycle. Here an additional release mechanism, the pawl release lever, was provided.

The pawl release lever (detail K591) is loosely mounted on the ratchet sleeve with clearance between the swinging arm and the ratchet gear. Existing space was sufficient to accommodate the lever without modification to the layout. The operating ramp lifts the pawl out of engagement when the lever is operated. The motion is constrained by a pin in the swinging arm which enters a travelling slot in the lever. During loading the direction of rotation (clockwise) biasses the lever to hold the pawl in the disengaged position .

Paper is wound onto the feed roller from the magazine by turning the balanced handle rotates the roller. The paper is cut by drawing a blade across a vee–notch cut into the upper spacer on the magazine.

The paper loading sequence is as follows:

1. Halt cycle with printing mechanism lowered as far as possible without engaging the feed pawl.

2. Attach loaded magazine to printing mechanism by dropping into slots.

3. Thread paper past upper spacer, over loose roller on magazine, and under paper support bar.

4. Clamp end to feed roller using cross clamp.

5. Lift backstop pawl by turning release lever clockwise and wind paper onto feed roller.

5 Release backstop pawl by returning release lever to home position.

6 Cut paper along length of upper spacer and feed around paper support bar.

7. Tension the take–up spool by rotating one turn clockwise and secure end using cross clamp.

8. Remove magazine.

*Counterbalancing Vertical Racks*

Each of the vertical racks has the run of teeth for the mating pinion displaced by the vertical pitch of the figure wheels along a steep diagonal (174). Though the effective section of gearing does not extend the full length of the racks all thirty vertical racks are assumed to be the same length and run the full height of the engine. (The only view of this is on 174 which shows the lower section only.) The racks are made of steel and their collective weight is 41.4 lbs. The racks, which are in loose contact over their length and will be lightly coupled through sliding friction. The effect of this is increased by the L–shaped lapping of adjacent pairs of racks (173).

There is no evidence in the original design for counterbalancing the deadweight of the racks and there are two points in the cycle where there is a risk of runaway i.e. of uncontrolled downward motion as the racks lower themselves under their own weight. The first possible runaway condition occurs during the transfer from the last even sector column to the figure racks. The transfer from odd to even columns in the first half–cycle sets the tabular value on the eighth column. The tabular value is transferred from last even sector column to the figure racks (176) by the last even sectors during the second half–cycle i.e. during even to odd addition. The second half–cycle commences with the tabular figure wheels in general displaced anticlockwise from zero, even sectors fully raised (disengaged) and figure wheels locked. The even sectors then fully lower to engage, figure wheel locks withdraw, and figure wheel axes and zero stops lower. The figure racks are unlocked throughout this sequence. The expected action here is that the figure wheel axis internal arms will drive the figure wheels to zero, and in so doing drive the figure racks via the sector wheels in a controlled manner. However, when the figure wheel locks withdraw there is a danger that the racks lower under their own weight and run ahead of the figure wheel drive arms. The effect is for the vertical racks to drive the train (figure racks, sectors and figure wheels) until the outer nibs of the figure wheels ram against the zero stops. The worst–case condition occurs with all figure wheels set at '9'. Here all racks fall the full vertical travel of 2.827 inches. The trains for figure wheels at positions intermediate between '0' and '9' will be driven in proportion to the figure wheel value; figure wheels set at zero are unaffected.

The coupling of racks through light sliding contact will have two effects: one effect is for less mobile racks to act as a brake for runaway neighbours; the second effect is to avalanche process in which the less mobile racks join the downward motion through lateral coupling.

The runaway does not introduce either calculation or printing errors. The effect is simply to set the figure racks up with the desired tabular value in an uncontrolled rather than controlled manner, and to impact the zero stops.

The second risk of runaway occurs during the restoration after stereotyping. Once the tabular value is transferred to the printing/stereotyping heads the figure rack locks and figure wheel locks secure the train and immobilise the mechanism during printing and stereotyping (385a). The locks then disengage and the intended next action is for the sweeping sector arm to engage with the pegs on the upper side of the sectors to restore the tabular value, return the even sectors to zero, and restore the figure racks. It is at this point that the second runaway condition is liable to occur. The general condition is for the sectors and figure racks to be displaced by different amounts from zero. With the locks removed and the figure wheel zero stops raised, the racks will tend to lower again under their own weight and drive the train in reverse. The limit of the motion this time is not the figure wheel zero stops but the advancing sector wheel restoring arm which will impact the drive pegs.

Neither of the two runaway conditions introduce calculating or printing errors. In the both cases the restoring action of the drive arms will complete correctly. However, the ramming of the mechanism against the zero stops in the first case, and against the rotating sector restoring arm in the second, is regarded as undesirable. There is also concern about the load of the racks on the sector restoring arms when the racks are lifted after printing as well as the overall load on the manual drive when the racks are collectively lifted and restored to zero.

To avoid the risk of runaway the racks are counterbalanced by providing individual counterweights for each rack. 174 shows 30 racks 1/8" wide and spindles nominally 1/4" wide. It is evident from the plan view (173) the arrangement is more subtle: the L–shaped lapping discussed earlier means that the 15 odd–numbered racks face the front (digits 1,3,5, ... , 29) (the least significant digit represented on the right–most rack), and the 15 even–numbered racks face the rear. The L–shapes provide a vertical face 1/4" wide for the rack teeth and for meshing with the 1/4" pinions. 174 has solid lines at 1/8" pitch to represent the racking members and the section with the racking teeth 1/4" wide. This indicates that the only sections that are lapped are the toothed section of each racking member; if the lapping was extended beyond the toothed section the alternate vertical lines would be broken. However, the use of broken lines to show hidden edges is not consistent elsewhere, and drawings are sometimes an inconsistent mix of sectioned and diagrammatic views. Licence was therefore taken to extend the lapping beyond the toothed sections upward and downward but clearly stopping short at the lower end where the printing sector racks revert to a true 1/8" pitch i.e. with no lapping flange. Details of the racks are shown on K332 A–Q.

Contemporary practice indicated that if the racks were to be counterbalanced this would likely to have been achieved by individual weights suspended by cords wound on drums in the pinion shafts. Two factors weighed against this arrangement: the lowermost pinion shafts have insufficient clearance for the full downward travel of the weights; and the counterbalancing load is transmitted through the friction of the shaft bearing and pinion gearing, and this force is variable. The arrangement preferred involves individual counterweights being attached directly to each rack by a cord suspended over a pulley on a shaft above. The weights are long steel strips (8" x 1 1/2" x 1/2") and weigh 1.38 lbs each. Each of two pulley shafts run left to right (front and rear) carry 15 3–inch diameter phosphor bronze pulleys (K322) 1/4" wide. Viewed from the left hand end of the engine the cords are centre–parted evenly to the front and rear.

Adjacent weights are in light contact and inhibit any tendency to spin. The weights are suspended in two tiers i.e. 15 weights in two tiers at the front (eight above and seven below) and 15 in two tiers at the rear. Channels cut down the faces in contact allow paths for the cords to the lower tiers. The arrangement is shown in K22 and K21. The top of the racks are cut away and drilled with 1/6" holes for the cords. The purpose of the cut away is to ensure central suspension points to avoid listing off the vertical.

There was concern that the cords would slacken momentarily when the racks were raised by the pinions, with the danger that the cords would jump the pulley wheels. A brief calculation reassures otherwise. For a ten–digit restoration of the rack the rack is raised 2.827" in 60° of the cycle. In the first 0.01 sec the free–fall distance of the weight is 0.019" and the rack movement is 0.028" i.e. 0.009" cord slack. The pulley channel depth is approximately 0.0625" i.e. the initial slack is insufficient for the cord to jump the pulley.

An additional benefit of counterbalancing is the reduction of wear on the figure racks locks and figure wheel locks. As the engine wears, the corrective action of the locks increases. Without counterbalancing, the locks would be inserted against the weight of the racks. This additional load would accelerate wear of the locks as well as increase the shock–load on the cam. This is regarded as undesirable given the steep pressure angle of the lock cam profile and the inability to turn the engine without counterbalancing of the locks with additional springs. The provision of counterbalances for the racks was considered to be consistent with the practice of providing spring counterbalances in the original design for the figure wheel shafts (163).

## Stereotype Table

*General*

Drawing References: BAB [A]**147**, 163, 164, **166**

The stereotype table is the apparatus for moving the trays containing soft material (papier mache ('flong'), or soft metal) into which results are impressed. The soft material is then used to make the plates from which tabulated results are printed. The printing and stereotyping apparatus is shown on 163 left. The printing apparatus is the box–like mechanism bolted to the left hand vertical frame members roughly half way up the engine. The stereotype table is the apparatus immediately below with the characteristic wish–bone struts stiffening the cast leg supports.

The result of each calculating cycle is set up on the small and large stereotyping sectors each with thirty type wheels. These are shown in detail on 172 and in outline at the top of 147. The two sets of stereotyping sectors are lowered together to impress the result into the material in the trays below to provide two sizes of type. The stereotyping sectors have vertical and rotational degrees of freedom only but have no freedom to move in the horizontal plane. The position of the result on the page is therefore determined not by lateral motion of the heads but the position of the tray below the heads. The stereotyping apparatus is in effect a programmable X–Y platform which repositions the trays under the stereotyping heads each calculating cycle to receive each new result.

The format of the tabulated results is programmable. The apparatus provides options for line–to–line printing in one or more columns, or column–to–column printing. The number of lines per page and line separation are also programmable. The various layout options are selected by choosing pattern wheels. One pattern wheel, selected from a choice of four, determines line–to–line spacing; a second pattern wheel, also selected from a set of four, controls column–to–column movement. The pattern wheels are shown on 163 and 164 extreme left, and in greater detail in 166.

The stereotyping table forms part of the control mechanism of the calculating drive. When a page is completed in the format preset by the pattern wheels, a weight is released into the cylindrical cup shown in 163 bottom left. The cup is connected to the scoop lever of the main drive clutch above the cam stack at the far right of the engine via a cable guided by pulleys. The clutch disengages the drive and the handle runs free. Uncoupling of the drive at the end of a page halts the engine without any overrun and ensures that the integrity of the figure wheel settings and control mechanism when the calculation run is resumed. The automatic halting of the engine at the end of a page is to allow the material in the trays to be renewed. This is done by fixing fresh trays, prepared with suitable material, on the carriage.

The stereotype apparatus is wider front to back than the calculating section (164) and the cast stand for the stereotyping table rests on the base plinth i.e. the legs are well clear of the long framing members supporting the vertical framing pieces.

*Description of Operation*

The material receiving the impressions from the stereotyping sectors is held in trays called matrix pans. Separate matrix pans for large and small type are shown at the top of 147. The matrix pans are fixed to a travelling frame or carriage (147 top right). The flong needs to harden before being removed from the pan. The pans therefore need to be removable to allow replacement pans, loaded with flong, to be fixed so that the calculation run can proceed while the impressed material is drying. No means of fixing the pans to the carriage is shown and no means of spacing the relative positions of large and small pans is indicated. However, the end view (147 top left) shows a flush fit along the left and right page edges and the Elevation shows the same top–of–page clearance for both small and large pans.
The position of the pans on the carriage is therefore fixed left–to–right by the width of the frame and the assumption is that the pans are wedged in place top–to–bottom (page–wise, front–to–rear as viewed from the front of the engine) in accordance with contemporary printing practice.

The carriage runs on two pairs of inverted v–shaped runners. One pair gives the freedom to traverse column–to–column on any given line ('v's shown in section in 147 End View); the other pair, at right angles to the first, gives the line–to–line freedom down the page ('v's shown in section in 147 Elevation). The two orthogonal sets of sliders provide the necessary degrees of freedom in the X–Y plane. The line–to–line motion of the pans corresponds to right–to–left motion of the carriage in the End View (147 (top right). Facing the engine this corresponds to a front–to–back motion i.e. the pan retreats as the table progresses down the page. The column–to–column motion corresponds to right–to–left motion of the carriage in the Elevation (147 top left) i.e. the carriage traverses right–to–left as viewed facing the front of the engine.

The carriage supporting the two matrix pans is shown driven in two separate ways. For stereotyping with large type the carriage is driven by a pair of racks and gears shown in the three views in 147. The two gears ($^5\boldsymbol{S}$, $^5\boldsymbol{P}$), keyed to drive shaft $^7\boldsymbol{A}$ engage with racks $^1\boldsymbol{R}_1$, $^1\boldsymbol{R}_2$ which are fixed to the underside of the carriage. Incremental rotation of the shaft provides the line–to–line advance of the platform. When stereotyping with small type, the carriage is driven by a single pinion $^5\boldsymbol{a}$ which engages with rack $^4\mathbf{R}$. The reduced pitch of the pinion ensures smaller line separation for the small type. No means of fixing the single central rack to the underside of the carriage is shown in either 147 or 163. This is an incompleteness in the drawing. The gears for the double rack and the pinion for the single rack are keyed to the same drive shaft. Since the carriage for both matrix pans is a single assembly it is clear that the drives are mutually exclusive: if both were engaged there would be a contention between the different pitches of the two incremental advances. It is clear from this that stereotyping in both small and large type simultaneously (i.e. in the same run) was not intended. Only one type can be used at a time and the drive for the unused size would need to be disengaged. Again, there is no indication of how this is to be done. The pan for the smaller type is shorter than the large pan by roughly the same factor as the difference in pitch between the pinion teeth and the pitch of the gear teeth. This indicates that the intention of the smaller type was not to produce more lines per page but to provide an alternative page size for the same run of results. The smaller type does however does provide the option of up to three–column layout compared to a maximum of two columns in the larger type.

The pinion is centred between the side frame members by two sleeves which act as spacers. The pinion and the sleeves have travelling keyways which slide on the long key fitted to the drive shaft. The whole assembly traverses during column–to–column motion (up–down motion in 147 plan). The pinion is shown wider in 163 than in the two views in 147. This is a drafting inconsistency.

The control mechanism for the stereotyping table is shown in the three views in 166. The two sets of pattern wheels ($^3$**N**, $^6$**N**) are shown as the large wheels immediately below the two horizontal bars in 166 (top left view). This right hand pattern wheel determines line–to–line motion ($^3$**N**); the left hand pattern wheel ($^6$**N**) determines column–to–column motion. The left and right hand pattern wheels jointly determine end–of–page action via the cluster of levers and catches positioned between them in the elevation in 166. The motive power for driving the pattern wheels during stereotyping is provided by weights ***A*** (column–to–column) and ***B*** (line–to–line) suspended over pulleys on the pattern wheel shafts (166 Plan and elevation). The intermittent release of the pattern wheels to advance the carriage and the end–of–page action are controlled by the two horizontal bars ***E*** and ***F*** (166). The drive to rewind the weights is derived from the main drive shaft (which rotates continuously during calculation) via a dog clutch shown as part of the circular assembly at the bottom centre of the end elevation in 166. The rotation of the line pattern wheel during stereotyping is discrete, with the size of the steps determined by the spacing of the teeth. The rewind action is a smooth anti–clockwise rotation.

The drive for the control mechanism derives from a pair of conjugate cams ($^4$***n***, $^4$***m***) which are driven directly from the main drive shaft, on the underside of the engine (163). The contact cam–followers $^5E_1$, $^5E_2$, and rocker shaft assembly converts the rotary motion of the cams into reciprocating motion of the lever $^5E_3$ (166 elevation). The rounded head of the lever is trapped in a recess in the lower oscillating bar (***E***) and the bar executes one linear reciprocating oscillation each calculating cycle. The rise and fall of the cam profiles occupies 29° of the 360° cycle (13° each way and a 3° dwell) i.e. about 12% of the cycle. With an average cycle time of six seconds, one outward and return excursion (13/16" stroke) of the oscillating bar takes about 0.7 sec. The bar is therefore stationary (to the right) for most of the cycle.

The incremental intermittent motion for line–to–line motion is produced by the escapement action of the oscillator bar and line catch ($^1$***B***) on the line pattern wheel. A peg in the lower oscillating bar operates the line bar lever during the outward (right to left) excursion of the bar. This releases the line catch which frees the pattern wheel to rotate. The pattern wheel is driven clockwise by the suspended weight, ***B***, and rotates by one tooth for each release.

The restoring force for the line catch is provided by a leaf spring fixed to the back plate and bearing on a lug on the line catch shaft (shown as dotted square just below notation '6**B**' in 166 plan). The motion of the pattern wheel steps one position clockwise each cycle. This is transmitted to the carriage drive shaft $^{7}$**A** (147) via a gear train. One gear ($^{3}$**B**) is mounted on the pattern wheel shaft (plan view 166). The second gear is shown in outline only drawn concentric with the line catch pivot in the end view of 166.

The line–to–line height is determined by the pitch of the pattern wheel teeth for which no detail is provided. Four line pattern wheels are shown permanently mounted on the pattern wheel shaft. Selection is made by sliding the line catch to engage with the required pattern wheel. The selection is secured by a screw which fixes the position of the line catch (166 plan). (The pivot shaft for the line catch obscure the pattern wheels which should be shown dotted in the plan view in 166. The pattern wheels are, however, depicted with solid lines. This is one of several examples of casual drafting convention.)

When the line catch releases the pattern wheel there is a danger of the pattern wheel running away under the influence of the suspended weight. As drawn, the pattern wheel would advance one tooth for each release if and only if the catch returns before a second tooth passes i.e. the correct operation of the drive relies on the machine being driven at the correct uniform speed to ensure that the catch returns in time to prevent the passage of more than one tooth. This is an unnecessary constraint and one difficult to guarantee. There is also the risk of runaway if the machine is halted with the line catch withdrawn.
It was regarded as inadvisable to rely on the constancy and continuity of the engine speed for correct line–to–line operation and it is unlikely that this was intended given the lengths gone to elsewhere to secure particular functions. The omission of a means of ensuring single–increment advance of the line pattern wheel was taken as a design omission requiring remedial action (see below).

*End of Column Action*

Stereotyping progresses down the page with successive releases of the line–to–line pattern wheel . The line rocker pivot which is integral with the oscillating bar moves to and fro carrying with it the line rocker $^{1}$**B**. The lower bar has a short section widened to accommodate a through–slot and the line rocker lever passes through the lower bar to engage a slot in the upper bar. On the outward stroke of the lower bar (right to left in the end elevation 166) the pad of the rocker lever is pressed against the right hand wall of the through–slot and drives the upper bar to follow the outward stroke. During the outward stroke the rocker lever is trapped at the angle drawn, and upper and lower bars and rocker assembly moves as a whole 13/16" to the left. On the return stroke the lower bar moves to the right. The top end of the line rocker lever is held by the upper bar and the line rocker turns anti–clockwise on the moving pivot leaving the upper bar stationary. This continues until the right hand foot of the line rocker meets a fixed end stop which prevents the foot travelling further to the right. (The end stop is shown as a zigzag series of right angles in the plan view of 166.) With the foot of the line rocker obstructed the line rocker lever drives the upper bar to the right as the lower bar returns and the line rocker returns to the inclined position shown. The overall effect is that the two bars execute the outward stroke together with the upper bar lagging the lower bar during the return stroke. The length of the strokes is the same (13/16") for both bars.

Integral with the pattern wheel is a line pattern arm ($^{3}\boldsymbol{D}$) shown clearly in 166 plan. The action described continues with the pattern arm stepping its way clockwise with the pattern wheel until the pattern arm approaches the nine o'clock position. The line rocker is displaced to the left when the pattern wheel is next released and the lug of the pattern arm fouls the foot of the line rocker. This holds the pattern arm just off the nine o'clock position. On the return stroke of the lower bar the foot of the rocker arm clears the lug and the pattern arm advances to nine o'clock with the foot of the rocker lever now trapped between the fixed stop to the right and the lug of the pattern arm to the left as shown on 166 plan and elevation. This last pattern wheel movement positions the matrix pan to receive the last stereotyped line of the column. With the foot of the line rocker trapped in its right–most position the next outward stroke of the lower bar drives the upper bar beyond the 13/16" travel of its own stroke by the lever action of the line rocker turning on the moving pivot. The additional throw of the upper bar has two effects: (1) the ramp on the upper bar (166 plan) operates the clutch lever, $^{5}\boldsymbol{B}^{2}$, which engages the clutch to rewind the line pattern wheel and thereby reposition the matrix pan at top of page; (2) the fixed pin on the left of the upper bar (as seen in end view 166) releases the catch for the column pattern wheel which drives the carriage across the page to the new column position. These two actions position the matrix pan to receive the next stereotyped result as the first entry of the new column.

The column pattern wheel ($^6$***N***) has two or three teeth depending on the number of columns required, and their distribution on the outer edge of the wheel is determined by the margin widths. In the case of multiple column stereotyping the extended throw of the upper bar that occurs at the end of a column will release the column pattern wheel to rotate one tooth pitch anti–clockwise as a result of the drive provided by the falling weight (***A***). Again there is no provision to prevent runaway in the event of incorrect engine speed or the engine cycle halting with the column catch disengaged. The rotation of the column pattern wheel drives the pan carriage laterally (right to left facing the engine) to position the pans for stereotyping the next column of results. The drive train from the pattern wheel to the carriage is via bevel gears shown in 163 and in outline in 147. The column pattern wheel directly drives the shaft $^6$***H*** (166 plan) which drives the cross–shaft $^7$***K*** via the bevel gears. The column–to–column drive train is completed by the racks and gears at each end of the frame shown in 147 End View.

The forked end of the clutch lever ($^5$***B***$^2$) drives the outer clutch bar ($^3$***T***) left to right as seen on Elevation 166. The clutch bar is keyed to he main drive shaft and rotates continuously during a calculation run. As the bar sweeps round, two lugs on the bar engage with two lugs on the stationary gear wheel ***T*** (166 Elevation). The lugs are 180° displaced and on different pitch circle radii so that the outer and inner lugs only engage with each other (two elevations of 166). The clutch bar drives the wheel ***T*** anti–clockwise (166 end view) which rewinds the line pattern wheel via gear $^4$***W***. Only one pattern wheel is automatically rewound: $^4$***W*** engages with either the line–to–line pattern wheel or with the column–to–column pattern wheel but not both. The selection is made by sliding $^4$***W*** along its shaft and dropping a latch into one of two slots in the spacing sleeve. The extra thickness of gearwheel ***T*** (Elevation 166) ensures that $^4$***W*** remains in engagement in either position. In the worst case of three–column format (small type only) the column pattern wheel has sufficient rotational reserve to drive the carriage across the page without rewinding. In the case of line–to–line stereotyping in multiple columns, the column pattern wheel is rewound by hand. In the case of column–to–column stereotyping, the column pattern wheel is rewound automatically and the line–to–line pattern is rewound by hand at the end of page.

The clutch is thrown out of engagement after about 340° rotation by two fixed ramps $^6$***a***, $^6$***c***, engaging with mating ramps on the clutch bar. The ramp on the upper oscillating bar, which originally activated the clutch lever, has long since withdrawn and the clutch lever is driven back to the disengaged position. The line pattern wheel is held in the rewound position, with the weight raised, by the line catch which rides over the teeth during rewind, with a leaf spring providing the return force. The line pattern wheel rotates less than one full revolution in driving the matrix pans from top of page to bottom. When the pattern arm is at the start position i.e. when the pan is at top of page, the arm is therefore clear of the nine o'clock position and spurious end–of–column action is avoided during a page run since the arm only reaches nine o'clock at the end of a column. The clutch bar rotates continuously during calculation i.e. its motion is not interrupted by engagement or disengagement.

The column pattern wheel pulley has sufficient reserve of cord to drive the carriage fully across a page without rewinding; the line pattern wheel pulley has sufficient cord reserve to drive the carriage down the page once without rewind. In single or multiple columns in line–to–line or column–to–column format a full page of results can be produced without interruption for manual rewinding . In the case of single–column stereotyping no manual rewind is required at all even at end of page: the line pattern wheel is rewound automatically and the column pattern wheel is never released to rotate since no left–to–right traverse across the page is needed. Any column pattern wheel can therefore be used for single–column operation as the pattern arm rather than the pattern wheel is the effective component. Once set at three o'clock at the start of page, the column pattern arm remains in this position to trigger the end–of–page sequence. For column–to–column stereotyping, the column pattern wheel is rewound automatically at the end of each linnd the line pattern wheel is rewound by hand at the end of a page. Both manual rewinds occur with the engine halted after a page is complete.

*End of Page*

If the page is incomplete when stereotyping reaches the end of a column, the control mechanism traverses the carriage to the next column and rewinds the carriage to top of page as described. If the page is complete at the end of column then the control mechanism halts the machine to allow the pans to be replaced. The end of *column* condition is indicated by the line pattern arm reaching the three–o'clock position.

This initiates the release of the column catch as described. In the case of multiple column stereotyping the column pattern wheel increments to the next tooth position and the pattern arm rotates towards the three–o'clock position; in the case of single column stereotyping the column pattern arm is set at nine o'clock at the start of page and remains stationary throughout. The end of *page* condition occurs when the line pattern arm is at nine o'clock signalling end of column and the column pattern arm is at three o'clock signalling the last column. The co–incidence of these two conditions initiates the end of page action via the cluster of levers and catches shown between the two pattern wheels on 166 end view.

The end–of–page action triggers the release of weight **C** (166 End View) suspended by a cord over stop pulley $^3\boldsymbol{L}_1$. The engine is halted by when the weight is released into the trough below (166). The falling weight operates the trip lever which primes the scoop cam to disengage the main drive clutch. The clutch breaks the drive between the operator's handle and the bevel gear of the cam drive shaft (see Section D).

The stop pulley is controlled by two radial levers which are integral with the boss of the stop pulley ($^3\boldsymbol{L}_2$, $^3\boldsymbol{L}_2$). These are shown between nine and ten o'clock and between ten and eleven o'clock in 166 End View. One lever is the column stop catch, the other is the line stop catch. Each of the levers is obstructed from clockwise motion by a trip lever ($\boldsymbol{T}_1$, $^2\boldsymbol{T}_1$) which rocks on a fixed pivot. *Both* trips need to be released to release the stop pulley: releasing one and not the other has no effect i.e. the arrangement is a mechanical AND gate requiring both release conditions to be met. The line and column trips are operated by extra arms integral with the line and column rockers respectively. The arm on the line rocker is shown at roughly nine o'clock (166 End View) and that on the column rocker at three o'clock.

The end of column condition occurs when the line pattern arm reaches nine o'clock; the end of page condition occurs when the line pattern arm reaches nine o'clock and the column pattern arm reaches three o'clock i.e. the end–of–page condition occurs when the end–of column and last–column conditions are both met at the same time. If the line pattern arm is at nine o'clock the line trip is operated during the outward stroke of the lower bar by driving the line rocker anticlockwise. The tilt of the line rocker releases the line stop catch. Similarly, if the column pattern arm is at three o'clock the column rocker operates the column trip to release the column stop catch. Only if both these conditions are satisfied is the stop pulley released. If either pattern arm is not in its terminal position (for example, end of first column in a multiple column format) then only the opposite trip is operated with no net effect on the stop pulley.

The trips operate singly when one or the other, but not both, of the end–of–column and last–column conditions are met. The trips also operate singly, and therefore without effect, during the return stroke of the lower bar when neither terminal condition applies i.e. while intermediate results are being stereotyped. It was observed earlier that while the upper and lower bars execute the outward stroke together, the upper bar lags the lower bar during the return stroke. During the interval of hysteresis the line rocker (in the line–to–line configuration shown in 166) tilts anticlockwise as the lower bar executes its return stroke and the upper bar remains stationary. This tilt is sufficient for the line rocker to release the line trip. The column trip is unaffected and the single release, which occurs with each oscillation of the bar, does not release the stop pulley. In the column–to–column configuration (see below) the column trip is operated each time the lower bar executes the return stroke again with no net effect.

*Conversion for column–to–column Stereotyping*

The configuration of the mechanism shown in 166 is for line–to–line printing in single or multiple columns. In this format each result is impressed below its predecessor to generate an output column. At the end of a full column, wind–back to the top–of–page occurs to restart a new page in the case of single–column operation, or to restart a new column at the top of the same page in the case of multiple column operation. For column–to–column stereotyping the carriage needs to traverse right to left (facing the engine) to accept successive results on the same line, then advance the carriage to the next line, and finally wind the carriage back left to right to start a new line. For two–column stereotyping the carriage is driven left and right across the page alternately and line increments occur after each pair of results; for three column stereotyping (option with small type only) the carriage traverses twice to the left, advances front–to–back by one line increment every third result, and rewinds fully to the right for start of new line.

The conversion from line–to–line column–to–column operation is made by removing the upper of the two oscillating bars and refitting it in an inverted position. Four pairs of vertical guides, integral with the lower bar, constrain the upper bar in the horizontal plane.
The upper bar is freed by removing the keeper fitted across one pair of guides shown second from the right in 166 end view. Once the bar is inverted the keeper is replaced. The two lugs on the upper bar (?v) which hold captive the upper end of the line rocker in the line–to–line configuration, face upwards in the inverted position, and are inactive. Two corresponding lugs to the left (166 end view) on the upper bar become active by trapping

the end of the column rocker instead. The conversion is completed by transferring the removable pin on the lower bar (166) to a corresponding position on the left of the bar so as to operate the column catch instead of the line catch. In line–to–line mode the fixed pin at the right hand end of the upper bar faces into the page and is inactive. In the inverted position this pin faces out of the page and operates the line catch. Finally, the rewind wheel, $^{4}\boldsymbol{W}$, is repositioned and fixed with the drop catch so as to disengage from the line pattern wheel and drive the column pattern wheel via idler $^{2}\boldsymbol{M}$ instead.

In the column–to–column configuration the column catch is released each cycle on the outward stroke of the lower bar. The release of the column pattern wheel advances the carriage to the next column via the drive train consisting of the shaft $^{6}\boldsymbol{H}$ (166 plan), bevel gears ($^{1}\boldsymbol{L}$, $^{6}\boldsymbol{L}$) (163), cross shaft $^{1}\boldsymbol{K}$ (147 End View), and gears ($^{1}\boldsymbol{N}$, $^{1}\boldsymbol{H}$) and racks ($^{2}\underline{\boldsymbol{u}}_{1}$, $^{2}\underline{\boldsymbol{u}}_{2}$) (147 End View). The upper bar, driven now by the column rocker rather than the line rocker, follows the lower bar on the outward stroke and the 13/16" travel is insufficient to operate the line catch. On the return stroke the upper bar lags the lower bar as before. The column rocker operates the column trip during the return stroke due to the tilt of the rocker arm during the hysteresis gap but unless the end–of–column condition is met (line pattern arm at nine o'clock) this has no effect.

In the case of two–column stereotyping the column pattern arm reaches the three o'clock position after only one release and the arm will therefore reach the three o'clock position every second cycle. As the arm is driven anticlockwise by the suspended weight (A) fouls the foot of the column rocker and the arm is held just off the three o'clock position. On the return stroke the foot of the rocker clears the pattern arm and the arm nudges home to three o'clock. The carriage is now positioned to receive an impressed result in the second column. On the next outward stroke the pattern arm obstructs the foot of the column rocker and the rocker levers the upper bar to the left (166) in excess of the standard 13/16" travel. The extended throw of the upper bar allows the fixed pin to in the upper bar to operate the line catch and the release of the line pattern wheel advances the carriage by one line increment. The extended throw also allows the ramp on the upper bar (166 plan) to operate the clutch lever ($^{5}\boldsymbol{B}^{2}$) and engage the clutch which rewinds the column pattern wheel as described earlier.

The sequence described is repeated with the a line increment occurring after alternate pattern wheel releases. The line pattern wheel advances clockwise until the line pattern arm reaches nine o'clock. The end–of–page condition is met when both pattern arms are horizontal and facing each other. On the next outward stroke the two trips are released by the rockers. This releases the two stop catches and the stop pulley releases the weight to halt the calculation run. For column–to–column stereotyping the column pattern wheel is rewound automatically, and the line pattern wheel and stop pulley are rewound by hand as part of the end–of–page procedure. For three–column stereotyping (small type only) the column pattern wheel has three teeth and the line pattern wheel is released after every two releases of the column pattern wheel.

*Changing Pans*

The line–to–line movement of the carriage is derived from the single drive shaft, $^{7}\mathbf{A}$ (147, all views). The line advance is driven by the central rack and pinion when stereotyping using small type, and by the rack and pinion pairs in the case of large type. It was observed earlier that the pitch of the line–to–line advance when using small type is less than that for the large type and that since the carriage is shown as a single assembly, a given run of stereotyping would use the small matrix pan or the large matrix pan but not both. 147 End View shows the small and large pans aligned at top–of–page below the type heads with the common starting positions separated by 17.25" — this a rare absolute dimension.

The type heads need to impress the soft filler and the depth of impression will require the type heads to penetrate into the body of the flong. To prevent the type heads on either side of the active heads fouling the lip of the pan at the top and bottom of the page the pan is filled so that the upper surface of the flong is flush with the lip of the pan. When stereotyping using small type the end–of–page position of the pan leaves the pan run out to the left (147 End View) with the type heads above the last line of the page; the large pan (without soft filler) will be partially advanced down the page. The automatic halting of the engine at the end–of–page and the carriage rewind that ensues prevents stereotyping overrunning the pan and the type heads are thereby prevented from fouling the right hand lip of the small pan (147 End View). In the case of stereotyping with small type, the large pan (unfilled) can therefore be present on the carriage without penalty. However, the inverse is not true: when stereotyping with large type, as the carriage traverses down the page the small stereotype heads will foul the lip of the small matrix pan as the end of the small page is overrun. The small matrix pan would therefore need to be removed from the carriage when stereotyping with large type and a spacer inserted in its stead to secure the large pan. The Timing Diagram ([F] 385/1a) shows the table clutch (column 29) engaging during interval 42 (Babbage cycle division corresponding to 302.4°) and the engine halting under automatic control at. Does this mean that the events between 350° and 360° occur after restart?] following the release of the stop pulley and weight **C** The automatic carriage rewind commences with clutch engagement so that by the time the engine halts at the end of cycle, the carriage is already partially rewound. Rewinding continues throughout the first half of the next cycle and is completed during interval 32 (230.4°) i.e. well into the second half–cycle.

In the case of column–to–column operation, the pans will halt with the last–line position under the type heads and the carriage partially returned towards the first column (left to right facing the engine). For small type stereotyping the small pan will be run out to the left (147 End View) largely clear of the type heads and can be removed, and if necessary renewed, with relative ease. Removal of the large pan, however, is obstructed by both sets of type heads. For large type column–to–column operation this difficulty can be overcome by re–engaging the main drive clutch after automatic disengagement and advancing to the next cycle and halting the machine manually during interval 32 (230.4°). This corresponds to the completion of the automatic carriage rewind cycle and coincides with the disengagement of the table clutch. In this position the large pan is fully run out to the right (147 End View) with the type heads positioned at the start of page but with the next result not yet impressed. Removal and replacement of the pan is relatively unobstructed in this position. The automatic halting of the engine serves here as a timing marker rather than positioning the pans for removal.

In the case of line–to–line operation the carriage will be partially rewound back down the page (front–to–back facing the engine) when the machine halts automatically. In line–to–line mode the removal and replacement of small or large pans will be partially obstructed by the type heads. For small pan operation the solution is to remove the spacers and wedges positioning the pan and to move the pan towards the gap between the type heads (left to right 147 End View). This will allow it to be slid out. For large pan operation the solution is as described for column–to–column operation i.e. to re–engage the main drive clutch and advance the cycle until the carriage is rewound to top–of–page, the position shown in 147 End View.

**Implementation**

*Runaway*

The line and column pattern wheels are driven by weights, ***B*** and ***A*** respectively (166 end view) suspended over pulleys $^3\mathbf{G}$, $^6\mathbf{G}$ fixed to the pattern wheel shafts. When the line or column catch is released by the action of the pegs on the oscillating bar the line and column pattern wheels a free to rotate driven by the appropriate suspended weight. However, unless the catches are is reengaged in time to allow only one pattern wheel tooth to pass (this is difficult if not impossible to ensure as the period of disengagement depends on the engine speed which, since manually driven, will be variable) there is the danger of the pattern wheels running under the influence of the falling weights.

Column 24 of the Timing Diagram (([F]385/1a) indicates the active period of line and column catch disengagement and the catches are referred to as 'Detent $^6\mathbf{C}_2$ or $^5\mathbf{C}_2$' at the head of the column. The function of the catches is to allow the pattern wheels to advance one tooth at a time in one direction, and be overridden during rewind in the other. Their function is therefore somewhere between an escapement and a ratchet and referring to the catches as detents, which usually implies that they can be overridden in either direction, is curious.

A modification is required to provide the escapement action i.e. to restrict the pattern wheel advance to one tooth increment only for each release of the line catch. The solution chosen is to retain the first line catch unmodified from the original and to provide a second line catch to prevent unwanted advance after each release. The modified arrangement is shown on GA L24. The pattern wheel and two line catches act as a ratchet and double pawl. Both line catches must act on the same tooth as the pattern wheel teeth are not continuous but are bunched in interrupted groups of five corresponding to five–line groups of print separated by blank rows for ease of reading. An additional peg in the lower oscillating bar is provided to operate the second line catch. During the rest period the second line catch is held out. The catch is released by the peg during the outward and is driven into engagement by the leaf spring; the second catch engages before the first catch is withdrawn. During the return stroke, the first catch engages before the second catch is withdrawn.

The guides for the oscillating bars require modification to accommodate the additional pegs. The original drawings show the lower bar supported at each end by channels in the two end supports (166 Elevation and end view), and the upper bar supported by four sets of guides

(**E**) attached to the lower bar.]. To allow space for the extra pegs in the lower bar, the outer two sets of guides attached to the lower bar are combined with the end supports to provide support for the upper and lower bars (L471, L472); the central two remaining guides remain unchanged.

The modification of the column catch is more extreme though the principle of providing a second pawl is identical. The column pattern wheel assembly is crowded more closely to the end framing pieces than the corresponding line pattern wheel assembly and there is insufficient space to accommodate a second column catch without modification to the original catch. The modified arrangement is shown in L24. The original transmission is changed from direct lever action, to a lever–driven gear. This converts the operation of the catch from leading to trailing action as well as creating sufficient room for the second catch. The conversion to trailing action has the additional elegance of restoring symmetry with the modified line catch arrangement opposite. The pivot position of the peg–operated arm of the first catch is unaltered and additional pivots are provided for the pawl lever and second catch. The second catch is also geared and is driven by additional pegs in the lower oscillating bar. The phasing of the two catches is identical to that of the line catches i.e. the second line catch is released for engagement before the first catch is withdrawn, and withdrawn after the first catch re–engages.

The modification have minor consequences for the pawl profile and pattern wheel teeth. The rise of the tooth on the original line catch is shown hooked for ratchet action, and a sawtooth on the reverse side so that the catch can be overridden during rewind (166 End View); the original column catch has a different tooth profile (166 End View). The double pawl action makes the hooked tooth unnecessary and the fact of the modified column and line catches both being trailing levers allows all four pawls (line and column) have radial tooth rises on the active faces. Standardising the pawl profile also means that all pattern wheels (line and column) can have the same tooth profiles; the original arrangement of one leading and one trailing lever required different tooth profiles.

*Split Carriage*

The two elevations in 147 show a single platform bearing the two matrix pans mounted on inverted v–shaped runners. The carriage bearing the two pans tracks on the long runners to provide line–to–line motion (front–to–back motion facing the engine) (147 end and front elevations); the long runners track on transverse runners to provide column–to–column movement (right to left facing the engine). The drive for line–to–line motion is provided from shaft $^{7}\boldsymbol{A}$ via two rack and pinion arrangements. The larger line–pitch required for the larger typeface is provided by the pair of racks $^{1}\mathbf{R}_{1}$ and $^{1}\mathbf{R}_{2}$ fixed to the underside of the matrix pan platform engaging with pinions $^{5}\boldsymbol{S}$ and $^{5}\boldsymbol{P}$ respectively; the reduced line pitch for the smaller typeface is provided by the single central rack $^{4}\mathbf{R}$ and pinion $^{5}\boldsymbol{R}$ (working point $^{5}\boldsymbol{a}$).

While the general intention seems fairly clear there are mystifying aspects of the detail. 147 Plan and Elevation show an uninterrupted sleeve $^{4}\boldsymbol{B}$ on the drive shaft. The sleeve is shown keyed to the shaft by a sliding key along the full length of the shaft with the keyway extending along the full transverse travel of the carriage – this so as to provide line–to–line drive while the carriage traverses column to column. The End View shows a key cut into the shaft. It is unclear (depending on where the section is considered to be taken) whether the pinions $^{5}\boldsymbol{S}$ and $^{5}\boldsymbol{P}$, or the sleeve, are keyed to the shaft. If the pinions are keyed to the shaft then the sleeve acts as a spacer separating the pinions which are sandwiched by collars against the sleeve. On the other hand if the sleeve is keyed to the shaft then the pinions are integral with the sleeve and what would otherwise be outer collars become bosses which space the pinions from the frame.

The central single pinion $^{5}\boldsymbol{R}$ (with working point $^{5}\boldsymbol{a}$) is driven from the same shaft $^{7}\boldsymbol{A}$ but the method of coupling between the shaft and pinion is not clear. (The fact that the central pinion is driven by $^{7}\boldsymbol{A}$ is confirmed by the notation of trains in Spon between pages 251 and 252, 'Trains of the Printing Part of the Analytical Engine', bottom right). Dotted lines (Plan and Elevation) indicate that the pinion fits over the sleeve though no pinning or other means of coupling the pinion to the sleeve is shown. Similarly, no means of fixing or securing the single small rack is shown: this is drawn levitating freely in 147 end elevation.

The fact that the sleeve $^{4}\boldsymbol{B}$ has a different index of identity to the pinion ($^{5}\boldsymbol{R}$) indicates that they are separate pieces. However, the notation of trains indicates that shaft $^{7}\boldsymbol{A}$ *and* a component $^{5}\boldsymbol{B}$, loose in relation to each other (indicated by parentheses), are driven together i.e. the 'loose' relationship is the linear transverse motion, and the driven relationship is the rotational drive transmitted by the key. If the identity index of $\boldsymbol{B}$ in 147 is taken to be an error i.e. the sleeve is taken to be $^{5}\boldsymbol{B}$ instead of $^{4}\boldsymbol{B}$ then the pinion and boss are integral with the sleeve and this resolves the problem of the transmission of drive.

The use of dotted lines elsewhere in the drawings is not always consistent. If the dotted line showing the sleeve as continuous inside the central pinion is taken to be an error then the sleeve can be regarded as broken i.e. is in fact two sleeves acting as spacers to locate the central pinion in the correct position and to space off the two larger pinions. The central pinion would then be keyed directly to the shaft. It is unlikely that the sleeve is continuous and passes through the central pinion and boss: there is little wall thickness remaining (147 Plan and Elevation).

The pans are located left–to–right between the sidewalls of the carriage. The top–of–page end of the small pan, and bottom–of–page end of the large pan, register against the end–walls. No means of securing the matrix pans to the platform is shown and no means is shown of ensuring that the pans remain butted against the end–walls. If both pans are assumed to be fixed to the platform then the separate rack and pinion drives cannot be engaged at the same time since the line pitch increments for the larger and smaller types are different and the different rates of feed would result in a destructive contention between the two drives. When stereotyping using the large typeface, the small rack and pinion drive needs to be disengaged, and vice versa for small typeface stereotyping. No provision is made in the original drawings for the disengagement of the large racks and the lack of detail in the means of fixing the small rack, and the lack of indication of how the small pinion is driven leaves open the question of whether the small rack and pinion were intended to be capable of being disabled. Access to the underside of the carriage to disengage either of the drives is severely restricted and it is difficult to see how a convenient means of disengaging the drives could be devised. The function of the flanges on each side of the central pinion is not clear. They may be intended to act as guides if the rack is to be uncoupled and refitted. This notion gives tentative support to the supposition that only one of the two sets of racks are operative at any time. But the case for the flanges is marginal and their inclusion remains mystifying. If the method of uncoupling the drives involves disengaging the racks from the pinions it is essential that they are replaced in exactly the same mesh so as to ensure correct positioning under the stereotype heads as the pinions are phase–locked to the pattern wheel drive. If the racks are incorrectly remeshed the results would be displaced on the page; there is also the danger of the stereotype heads fouling the walls of the pans. The need to refit the racks exactly places an operational constraint on any means devised for uncoupling the drives.

The mutual exclusion of the two drives was first assumed to imply that simultaneous stereotyping using both large and small types was not intended. This conclusion was supported by the fact that if the small pan is present in the carriage when stereotyping to the large pan then the small stereotype heads would foul the trailing wall of the small pan (at the end–of–page end) as the carriage advanced down the page. The small pan could not therefore be present during large–type stereotyping without risk of damage. The difficulty of devising a means of uncoupling the drives and the operational difficulties associated with disabling one or another of the drives to transfer between large and small stereotyping led to splitting the carriage into two with one section for the small pan and one section for the large pan. Both drives remain permanently engaged and simultaneous stereotyping using both pans becomes possible. The rate of feed for the two pans are different and, as the stereotyping progresses, the large pan gains on the small pan. Because of the different rates of feed the front lip of the small pan does not pass under stereotype heads and there is no danger of fouling. A rare dimension in 147 End View shows the distance between the first lines of the small and large pans as 17¼ inches. Working back from the diameter of the pinions gives the maximum travel of the large pan as 14.75 inches, and 5.9 inches for the small pan i.e. a feed ratio of 5:2. Retaining the original pan sizes, pan separation at start, and pinion diameters, the pan separation at end of page is 0.1375 inches (see GA 337 L 25) – this from the common starting point determined by the position of the stereotype heads.

The decision to split the carriage was made in the light of the lack of appropriate detail in the drawings and the perceived difficulties in devising a convenient way of uncoupling one or another of the line–to–line drives. Subsequent examination of the notation lends some support to the notion that both pans were intended to operate simultaneously (i.e. that both drives were to be permanently engaged) and that the incompatibility of the line–to–line feed rates is resolved by the small pan, driven by the central rack, being free to slide on the carriage as the carriage (and large pan) are driven by the two outer racks. The decoding of the notation remains incomplete. However, the identifier for the central rack, $^{4}\mathbf{R}$ as well as the identifier for the small pan $^{7}\mathbf{K}$, are underscored three times while the large–pan rack identifiers are underscored only twice (147 Plan and End View). This might indicate that the small rack has an additional degree of linear freedom in relation to the larger rack. The Spon train notation of 1856 may well be significant here (bottom right). The notation indicates that the carriage **H** is free to move in relation to sliders **G**, and **G** is free to move in relation to frame **F** (**H(G(F)G)H**). The line–to–line motion of carriage **H** (and the large pan) is shown derived from racks $^{1}\mathbf{R}$. (The pans are also show acted on by the large stereotype sectors). The column–to–column motion of **H** is imparted by **G** driven by racks $^{2}\mathbf{U}$. However, the small pan $^7$**K** is shown as a driven piece by one (or both) of **H** and **G** *as well as* by rack $^4$**R** i.e. the line–to–line motion of the small pan is derived from rack $^4$**R** *and not* from carriage **H**, while the column–to–column motion of the small pan is transmitted by carriage **H** through working points ***u***. (Working points ***u*** are shown clearly in 147 End View.) This lends support to the possibility that the small pan is free in the carriage in the line–to–line direction and moves in the carriage relative to the large pan during stereotyping. The fact that no fixings are shown to attach the small pan to the carriage may be considered supporting evidence for independent line–to–line–motion. The lack of pan fixings (large and small) could be an omission of detail in both cases. Alternatively, it could indicate that no fixings are needed with the large pan being driven line–to–line by the front wall (i.e. at the front of the engine) of the carriage, and the small pan by the central rack with pan free to slide in the carriage. In both cases the column–to–column motion is transmitted by the side walls of the carriage.

In the final design the principle of using both the small and large pans moving at different rates is retained. However, instead of having the small pan sliding within the carriage, the carriage was split into two with the separate sections of the carriage driven at the same time by their reseptive rack and pinion drives. There are two main considerations favouring this solution: the rack providing the motion for the small pan can be fixed permanently to the underside of the carriage so resolving the operational difficulties associated with uncoupling and re–engaging the small carriage with its drive each time the pan needs renewal; secondly the carriage is longer than the pan alone and is therefore more stable on the v–slides slides.

The general arrangement of the new carriage is shown in GA 337 L25. Detail of the small pan rack is given in L352 (called 'rear' i.e. away from the front of the machine), and that of the large pan racks in L351A (LH) and L351B (RH). The rear pan is some distance away from the rack that drive it and the gap needs to be bridged. The rack is fixed to a ski–shaped support ribbed for stiffness in view of the long overhang. The top bearing surface of the support slides on the underside of the front carriage which acts as a guide. The rear carriage was extended to use more of the slide than shown for the single carriage. This provides a more stable guide for the rear carriage especially in view of the long overhang of the rack support. The new front and rear carriages are of equal size i.e. both carriages can accommodate the large matrix pan. Using small type and the large pan, reversing the pan at the end of page allows a maximum of two sets of three columns back–to–back in one pan.

Removal and replacement of the small pan is best carried out when the carriage reaches the end–of–page position i.e. when the engine stops under automatic control. To remove the large pan the engine cycle needs to be advanced until the carriage is wound back to the top–of–page position i.e. with the carriage at the extremity of its travel towards the front of the engine

The sleeve $^4$**B** shown as continuous on the shaft in 147 Plan was split into two. The two sections locate the central pinion by sandwiching it between them. Bosses were added to the large pinions and two outer collars are fixed to the shaft between the pinions and the side members of the frame. The full–length key was dispensed with and replaced instead by three short sections of key, one for each pinion, trapped between the two spacers in the case of the central pinion and between the spacers and the outer collars in the case of the pinion pair. The sleeve is not keyed to the shaft. The use of short sections of key is intended to reduce drag as the carriage traverses column–to–column.

*Column–to–Column Dashpot* [Ref: Red Book, page 130]

Babbage gives no details of the pattern wheel tooth layout or of the intended options for formatting the results on a page. In establishing appropriate layout combinations (see X23) taking account of the pan sizes and font heights the maximum line–to–line carriage movement emerges as 0.275 inches. This is the maximum distance the carriage will advance in one complete calculating cycle. The falling weight that drives this motion ('Weight ***B***' in 166 End View) needs to be large enough to overcome initial static friction without hesitation. If the weight is heavy enough to do this then the carriage will accelerate rapidly until stopped by the next pattern wheel tooth under direct uncushioned impact. With only 0.275 inches linear travel between stops the impact and jarring should be within acceptable limits. However, in the case of column–to–column motion the carriage will step as much 5¼ inches between stops. The action of the second safety pawl halts the initial rotation after 1/16 inches which still leaves a full 5 3/16 inches between stops. The uncushioned impact of a heavy carriage (now made heavier by splitting into two equal carriages) accelerating over approximately 5 inches was regarded as potentially damaging and a removable oil–filled dashpot was included.

The dashpot is fitted under the carriage along the centre line of the carriage frame **F** (147 Plan) in direct line with the boss of the large idler gear shown in 163 between the right hand side of the stereotype table frame and the left hand vertical frame piece of the calculating section. Detail of the dashpot assembly is given in L22. The dashpot acts as a linear speed governor. Its action is unidirectional. In the damped direction (right to left) the piston forces oil through an annular aperture. During the return stroke (left to right) the piston is driven against a spring and opens the oil–feed aperture to allow undamped movement. The rate of feed for a given grade of oil is adjusted by exchanging packing washers from one side of the conical plunger to the far end of the return spring so as to alter the size of the annular aperture. The arrangement ensures that the length of the return spring remains constant for different feed rate settings. Since the carriage is wound back during the return stroke the travel is controlled and damping was not considered necessary.

*Additional Fixing of the Stereotype Table*

The frame of the stereotype table is shown standing on rails which run the full length of the engine. Other than the common rails, no fixing is shown between the table assembly and the calculating section framing and this was considered risky. Additional fixing was provided by extending the boss of the idler wheel (163) to pass through a bar fixed between the two vertical framing members of the calculating section. The boss is secured on the far side of the bar by a collar. This arrangement allows the correct registration of the table to avoid front–to–back tilt. Jacking points on the table legs allow for adjustment and for correction of any tilt in both planes (front/back, left/right).

*Drop–height for Drive Weights*

Working back to the pattern wheel pulleys from the carriage shows that a pulley rotation of 340° is required for end–to–end travel line–to–line. The corresponding length of the drop for weights ***A*** and ***B*** for a 9 inch diameter pulley as indicated (166 end elevation) is 26.7 inches ($340/360 \times 9\ \pi$). However the drop from the underside of the weights to the floor is only about 11 inches. The extra drop is provided by additional pulleys suitably raised (L26). The pulleys are mounted on extension pieces fixed to the existing framework. Each of the masses ***A*** and ***B*** consist of ten separate weights. Each of the ten components is '**C**'–shaped i.e. a section of the ring missing so that they can be stacked and removed with a no dismantling. The most appropriate mass can be determined by trial to take account of friction, traction, load and eventually wear. Some form of flexible adjustment is required to cater for the presence or absence of one or another of the matrix pans.

*Pan Clamping*

No means of fixing the pans to the carriage is shown whether by screw fitting or wedges (147 End View and Elevation). The solution was to retain the left–to–right fixing as a slide fit of the pan into the machined tray in the carriage and to provide screw clamps to locate and fix the trays front–to–back (L25). Both trays have two screw clamps bearing on two wedge inserts. In the case of the small pan, two extension rods bridge the gap between the rear carriage wall and the edge of the pan (rods L363, and cross–sliding guide L354). The rods and sliding guides are removable loose pieces and the clamps are tightened with knurled nuts.

**Other Modifications/Additions**

*Pattern wheel arms*

166 (end view and Plan) shows a single lug at the end of each of the pattern wheel arms, which obstructs the foot of the line and column rockers when at 9–o'clock and 3–o'clock respectively. The lugs were modified to prevent the pattern wheel arms from advancing beyond 9– or 3–o'clock. The reasoning is fairly subtle and relates to the timing of the line catch release. In line–to–line mode when the line pattern arm reaches 9–o'clock (the position for printing the last line at end–of–page) the line catch is engaged with the last pattern wheel tooth. During the second half of the next cycle the last line is stereotyped with the pattern wheel arm still in the 9–o'clock position. On the next outward travel of the lower oscillating bar the line catch is released by a peg and the pattern wheel is held only by the pressure of the pattern arm on the foot of the line rocker. If this is insufficient to hold the wheel at 9–o'clock (as it might be even when the engine is operated without interruption), or if the engine is halted at this point by interrupting the rotation of the drive handle, then the wheel will advance a small distance until stopped by the safety pawl added to prevent runaway.

The line catch now rests on the untoothed portion of the wheel and cannot re–engage. The ramp on the oscillating bar operates the forked clutch lever which engages the clutch dogs. On the return stroke the lower oscillating bar disengages the runaway pawl. All that now stops the wheel running away is the engagement of the clutch. If the clutch engages successfully then a small runaway occurs and the dogs will clash as a result of backlash as they engage. A still less desirable situation might occur if the dogs clash when only partially engaged. If the clutch fails to engage through fault then we have a full runaway.

To prevent the pattern wheel arms rotating beyond the end–of–page position or last–column position (9 and 3–o'clock respectively) the pattern arm lugs were modified to form an 'L' with the horizontal foot of the 'L' engaging with the lower face of the rocker foot in both cases of line and column rockers.

*Stop Cord Pulley*

166 end view shows stop weight **C** deflected from the vertical at two points with no detail of the means by which the cord is to be routed. A cord guide was added at the upper point of deflection (detail L27).

*Stop Weight*

The geometry of the stop weight is uncertain. The depiction of the weights in 166 end elevation could be taken to be diagrammatic i.e. the simple rectangles not being representative of actual shapes. The only other depiction is in 163 bottom left. The completion trip is shown here strictly as a gutter though the geometry of the stop weight could be taken as a sphere or cylinder. It was first thought that the stop weight would be lifted clear of the sides of the completion trip during rewind. It would then be likely to twist on the cord under its own weight and fail to re–engage when next lowered. It was therefore assumed that the weight was intended to be a sphere and the trip a form of scoop. However, improved understanding of the action of the stop weight revealed that the weight would not be lifted clear of the gutter sides and the cylindrical geometry was retained.

*Resetting Handle*

The pattern wheels are conveniently exposed on the outside of the control mechanism for the stereotype table (163, 166 Plan and elevation) and these can be easily gripped for manual rewind. However, the stop pulley is well buried in the mechanism and finger access for rewinding is restricted. A handle was added to assist with the rewind. It is desirable to restrict the rotation of the stop pulley both clockwise and anticlockwise. Anticlockwise restriction would prevent the weight being inadvertently lifted above the sides of the completion trip during manual rewind, as well as ensuring that the catches and trips remain associated with the appropriate levers. Clockwise restriction is desirable to prevent the cord going slack and jumping the pulley sidewalls when the stop weight reaches the end of its travel after operating the completion trip. So a handle was provided at the outer end of the stop pulley shaft to assist rewind, and two stop pins added to restrict the handle rotation to 180° i.e. between 9– and 3– o'clock (337L24).

*Format Options* (Drwg. Ref: 337X230)

The original drawings give no details of formatting options either as page–layout descriptions or as pattern wheel profiles. 163 and 166 simply indicate a stack of four line pattern wheels and four column pattern wheels with no detail of pattern wheel variations. The format options (numbers of columns, margin widths, line pitch and line–group layouts) were determined from scratch and the pattern wheel tooth layout derived from these.

The known constraints are the size of the two matrix pans (147) which give the maximum length and width of the two stereotyping areas; the positions of the lines of stereotyping at top–of–page for a full page of results (147 End View); and the width of the column of print for each of the two type sizes (30 by 1/16 inches and 30 by 1/8 inches). (The number of digits in a result is not variable and is always 30).

Some format features can be inferred from these known constraints. For example, when stereotyping using both pans in multiple column format, the separation of the columns in the small pan is determined by the column width of the larger type. The width of a large–type column determines the left–to–right traverse of the carriage to position the pan for the next column. The margins between columns on the small pan are therefore wider than would be the case if the column jumps were optimised for the small type. The small pan can accommodate four columns of type but if the large pan is present during stereotyping the large stereotype heads would foul the left–hand edge of the pan at the start position. If the large pan was removed to allow four–column stereotyping then the drop weights would need to be adjusted to compensate for the lightened carriage. The decision was taken to restrict the maximum number of columns to three for small–type stereotyping. The maximum number of columns for large–pan stereotyping is limited to two by the pan width.

In the absence of more detail the following formatting features were taken as discretionary: side margins, margin widths between columns for multiple column formats, line pitch, and the numbers of lines grouped together before an additional line break (five–line groups were chosen throughout as consistent with the layout of contemporary printed tables).

The number of lines in a column or on a page, and the line pitch are chosen by selecting one of four line pattern wheels A, B, C or D. All remain permanently mounted on the end of the control mechanism. Selection is made by sliding the line catch to engage with the chosen pattern wheel (166, 337L409). Pattern wheel D is in fact blank spare deliberately left so to provide for a format combination not catered for by the other three wheels. The three line pattern wheels make provision for thirty, fifty or sixty lines per column — fifty for decimal increments, thirty and sixty for hexagessimal increments (trigonometric and astronomical tabulation) with the thirty–line format and large type used where large line spacing is required.

The arrangement of columns is determined by the four column pattern wheels, E, F, G, and H. Only one of the four is active at any time and the selection is made as before by sliding the column catch to engage with the chosen wheel. All four column wheels provide options i.e. there are no blank spares in the set of column wheels. The four wheels provide options for single, double and triple column printing, with various combinations of margin widths between columns. The specification of the pattern wheel layouts is shown in 337L409 A–D, and 337L408 E–H. (The hub for the pattern wheels is shown on L411).

Since automatic halting of the engine and automatic rewind is triggered by the end–of–page condition (line pattern arm at 9– o'clock, column pattern arm at 3– o'clock), this was taken as the fixed reference on the page and worked back from. A page–run therefore always ends in the same place with the last line stereotyped for a given pan always at the foot of the page regardless of the number of lines chosen (30, 50, or 60). Mapped back to the pattern wheel this means that the last tooth of each wheel is in the same angular position and the pattern wheels are in fact dowelled through at this position (337L409, L411). On the line–pattern wheel the position of the first tooth varies depending on the line pitch.

The same reasoning applies to the column pattern wheels. The column start position is wound back from the pan position for the last column (right–most on the printed page) and the last column is always positioned the same distance from the right–hand vertical edge of the pan. The smaller columns are centred on the larger columns because of the fixed relationship between the large and small stereotype heads (see 173). These constant right–hand margins are 3/4 inches for large type, and 1 11/16 inches for the small. For single column working these margin constraints still apply i.e. the single column will appear biassed to the right of the pan.

If the large pan is used in the position normally occupied by the small pan then the most compressed format can be achieved by stereotyping two pages of results in one pan. This occurs if two lots of three–column results, 30, 50 or 60 lines long are arranged back–to–back. This is achieved by removing the pan after the first page run and reversing it before replacing it on the carriage to receive the next run. THe resulting layout is shown in X23, bottom left). The bottom three views show examples of other layouts. The middle view, for example, shows a 2– column layout in small type withrow column margins.

The various combinations of line and column formats is summarised in the Summary Table in X23 (bottom right) with a few examples of the layouts in the diagrammatic views. The top half of X23 shows the two carriages with large and small matrix pans. The view at top right shows the large pan positioned to receive the first line of the first column. The centre view of the large pan shows the end position i.e. positioned for the last line, last column. The two views of the small pan correspond i.e. are depicted as they would be for simultaneous stereotyping with large and small type. The columns of large and small type are therefore aligned vertically with the spacing between columns determined by the width of the larger type. The gap between the carriages at the end–of–page is a constant 0.137 inches.

*Format Changes*

The procedure for restarting a run after automatic halting at end–of–page would be to rewind the column pattern wheel by hand (for line–to–line working) and reset it appropriately at the start position for either two or three column operation. For single column operation no column rewind is required as the pattern wheel arm remains in the 3–o'clock position throughout and any of the four column pattern wheels can be used. The position of the carriage when the engine halts under automatic control is the most convenient one at which to renew the small pan. The engine is then turned over and the carriage is wound back to top–of–page automatically. With the carriage rewound to the start position this is the most convenient time to renew the large pan as most of it is now clear of the large stereotype heads. For complete format change i.e. changing from line–to–line to column–to–column operation as well as line number and number of columns, reselect and set the new arrangement of the oscillating bars, reselect the appropriate line and column pattern wheels, and set to the start position by hand either the line pattern wheel (for column–to–column working) or column pattern wheel (for line–to–line working).

Whichever is not set by hand will be automatically rewound to start by the clutch. The selection of line or column pattern wheels for automatic rewind is made by sliding the rewind idler gear along its axis to mesh with one or the other as required.

*Variable Rewind*

In the original design the clutch–driven rewind returns the carriage from the end–of–page position to the top–of–page position i.e. rewind is always full–length. For fifty and sixty–line formats, which fill the page top to bottom, the full–page rewind is unproblematic. However, a potential problem arises for thirty–line working. Only about half the circumference of the thirty–line pattern wheel will have teeth (194° from first to last tooth, 377L409) with the remaining sector blank. Since a run always finishes with the carriage at the limits of its travel (back–to–front facing the engine), it is the start position of the run that varies and the thirty–line run will begin about half way down the pan. The first section of the pattern wheel will therefore be untoothed. The problem is that when the clutch dogs disengage at the end of rewinding to top–of–page, the carriage will run away until the line catch engages with the first tooth. A runaway of nearly half the full travel of the carriage would cause a damaging crash when the line catch finally engaged, and this should be avoided.

The thirty–line option is not specifically indicated in the original design and it could be argued that the runaway problem is a consequence of arbitrarily choosing a reduced page length. However, even if the original design was restricted to full–length formats the effects of fixed rewind still affects operation in column–to–column mode. Once again the last column position is the fixed reference and the start positions for two and three column formats differ. If separate column pattern wheels are dedicated for two and three column formats then a two–column pattern wheel would have roughly its first third blank and the problem of the column catch crashing into engagement on the first pattern wheel tooth persists. (The original design makes no provision for damping or governing thecolumn–column traverse of the carriage.)

If the amount of rewind could be made adjustable then the carriage could be rewound by the clutch to an appropriate start position and the potentially damaging situations avoided (both line–to–line and column–to–column modes). An additional advantage of an adjustable rewind is that the number of formatting options using four column pattern wheels can be increased.

Instead of dedicating separate pattern wheels for two and three column working, each of the four wheels could provide for full three–column working but only part rewound for two column working. (Any of the wheels can be used for single–column working as the pattern wheel does not move.) The column pattern wheels can then be used to provide a wider range of margin width options.

It is not evident from the drawings whether the original design was intended to cater for variable rewind. The end and front elevations in 166 show two rewind lugs $^2\mathbf{r}$, $^2\mathbf{t}$, but there is nothing to indicate whether the angular position of these lugs on the plate are variable so as to provide an adjustable amount of rewind. Reference to the drawings is inconclusive with respect to establishing the original intention of the design. To solve the problem of the start–of–page runaway in line–to–line mode in thirty–line format, and to provide the extra range of formatting options using the prescribed four column pattern wheels, provision was made to vary the amount of rewind derived from the clutch. Details of these arrangements are shown on L521 and L522.

The principle of the device is based on being able to alter the angular position of the rewind lugs in relation to the rewind gear. This is accomplished by mounting the two rewind lugs on a backplate the angular position of which can be varied in relation to the gear. The rewind gear is drilled with eighteen holes at 20° intervals and the plate is fixed to the rewind gear by two ring screws which pass through any of the pairs of diametrically opposite holes. To vary the amount of rewind, the ring screws are removed and the plate rotated until the engraved mark lines up with the pattern wheel letter on the rewind gear. Replacing the ring screws fixes the plate in the new position. (GAL27 gives a clear view of the clutch assembly). Twenty holes provide a series of fixed angular increments which reduce the start–of–page runaway to an acceptable level.

# Calculation

## General

This section describes the operation of the mechanisms that execute the addition of numbers according to the method of finite differences. The engine operates with vertical stacks of figure wheels thirty one digits high i.e. the maximum value of a number is thirty one figures. The least significant number position is at the bottom of the stack and the most significant position is at the top. Multi–digit numbers are represented by columns of figure wheels with one figure wheel for each digit–position of the number. During a calculating cycle 31–digit numbers on the figure wheel columns are added to 31–digit numbers on adjacent columns to the left in a process of repeated addition.

The column closest to the drive handle represents the seventh difference and the lower order differences are represented on the columns to the left as seen facing the engine. The column closest to the printer represents the result or tabular value. Babbage's notation identifies the $n^{th}$ difference column as $[Delta]^n$ and the tabular value column as ***T*** (163 eg). The highest order difference column (extreme right) is therefore labelled $Delta^7$ and successive lower differences are labelled with decreasing superscripts running right to left ending with the lowest order difference $Delta^1$ on the last but one figure wheel column on the left. The tabular value is held on the extreme left hand figure wheel column, ***T*** (see 161, 163, 164, 176 and 177). However, the convention adopted here is to number the columns from right to left 1 through 8 i.e. in the reverse sequence to the orders of difference. This number sequence follows the logical progression of information from right to left during a calculating cycle (see odd–even phasing below). With this convention there are four odd–numbered columns and four even–numbered columns and the tabular value appears on column 8 and the $7^{th}$ difference on the column 1.

For calculations requiring seven orders of difference, column 1 contains the constant difference. In situations where fewer than seven differences are used the active columns are those closest to the tabular value column, and columns closest to the drive handle are inactive. Columns are rendered inactive by setting all its figure wheels to zero. The inactive columns remain a functioning part of the calculation cycle but have no effect on the calculation. In circumstances in which fewer that seven differences are used the right–most active column (lowest numbered) contains the constant difference.

Using this numbering system the odd–numbered differences appear on the odd–numbered columns, and the even differences appear on the even–numbered columns regardless of how many differences are used in a calculation.

*Odd–even Phasing*

When using the method of finite differences as a manual technique for subtabulation it is usual to form the table in a way that allows addition of the higher order difference to the next lower order difference in a fixed incremental sequence. The manual calculation proceeds from the highest order difference (the constant difference) to the tabular value by progressing from right to left column by column. However, the sequence of additions as executed by the engine does not proceed in a stepwise way from right to left as one might expect from the manual method. One complete calculating cycle consists of two symmetrical half–cycles. During the first half–cycle the number values on the odd–numbered axes are simultaneously added to those of the even–numbered axes to the immediate left i.e. axes 1, 3, 5, 7 are added to 2, 4, 6,and 8 respectively. Similarly, during the second half–cycle all the even–numbered axes are simultaneously added to the odd–numbered axes again to the immediate left. Provided this is taken account of when setting up at the start of a run by offsetting the initial values on alternate axes by a half–cycle, the end result is the same i.e. each full calculating cycle results in a new tabular value which is the cumulative sum of the number of active differences.

The advantage of the phased operation of odd–to–even and even–to–odd half–cycles is that it allows a form of parallel operation which shortens the calculating cycle. Adding each column to its neighbour in a stepwise way would require eight distinct identical operations involving only two of the eight columns i.e. 75% of the engine would be idle at any one time. Simultaneously adding four columns to four columns in each of two half–cycles is more time efficient and also allows only 50% of the machine to be idle at any time. A further feature of this 'pipelining' effect is that the cycle time of the engine is independent of the number of differences used in the calculation. An additional advantage is the comparative simplicity of the control mechanism. A serial step–wise addition sequence would require a control system that independently activates each column pair in turn. This is substantially more complex a design than required for the phased pairing of alternate axes.

*Phases of the Addition Cycle: Giving off and Carry*

Each addition half–cycle is itself divided into two distinct phases. The first phase is 'giving off'. This refers to the process of transferring a multi–digit number on one of the columns, to the column of figure wheels immediately to the left via intermediate wheels called sectors. During giving off the number value on the right hand column is added to that on the left hand column. All thirty one digit positions in a column are transferred simultaneously. The addittion is effected in a tooth–by–tooth transfer in each digit position.

The second phase takes account of any carries. Each figure wheel has a spring–loaded warning device which is 'armed' if a figure wheel exceeds nine during giving off. This warns that a carry to the next decade is required for a correct result. Digit positions are polled in sequence starting with the least significant digit at the bottom of the column. If a warning is armed or set then the next figure wheel up is incremented by one position during the carry cycle. If the warning device is not set then no action is taken. Each position is polled in turn and Babbage calls the action 'successive carry'.

The mechanism caters for secondary carries. This situation arises when a figure wheel is set at 9 after the giving off phase and the next warning mechanism down is armed indicating that the 9 is to be incremented to 0 and a carry passed on. During the carry cycle the wheel set at 9 will be incremented to 0. This action will set the carry warning device in the next significant digit and the associated figure wheel will be incremented as the next action in the successive carry sequence. The worst–case carry condition arises if the column is set with nines in each digit position and a number between 1 and 9 is added to the least significant digit. In this case the carry ripples through the column as a series of secondary carries sometimes called 'domino carry'.

*Drawing References*

The key drawing is BAB [A] 171, particularly the section bottom left. The notation on 171 indicates that the view depicts the 1st and 2nd difference columns, [Delta]$^1$ and [Delta]$^2$ respectively (i.e. columns 6 and 7 in the new convention), with the intermediate even sector column ($\boldsymbol{R^2}$), odd warning axis ($\boldsymbol{W^1}$), and odd carry axis ($\boldsymbol{C^1}$).

In Babbage's design the relationship of figure wheels, sectors and carry mechanism is identical for odd and even axes. This can be seen in re is plan view shows the basic layout of a calculating unit viewed as a one–digit assembly that adds from right to left. This scale view is diagrammatic i.e it is strictly neither a section view nor a plan but a combination of the two: the figure wheel supports, locks, zero stops and shafts are shown in section but the carry axis is shown in plan as though with the top plate removed. The elevation shown in the upper part of 171 is also diagrammatic in the sense that mechanism has been opened out and the horizontal spacing has been expanded for clarity. Other drawings containing information on the calculation mechanism are 161, 163, 164, 176 and 177.

*Description of the Adding Mechanism*

The view on 171 lower left shows the basic mechanical unit of addition. The mechanism adds the 31–digit number on the right hand column to the 31–digit number on the left hand column taking account of any carries. The view shows two figure wheels coupled by an intermediate sector wheel (the partial wheel with a wedge cut away), a carry axis with helically arranged carry arms, and the warning and carry mechanism. Only the parts that are active in a two–column addition are shown. The carry mechanism for the right hand column is omitted as are the remaining six figure wheel, warning and carry axes. In the original design the arrangement shown in 171 is repeated symmetrically for the eight columns i.e. sector axes couple adjacent figure wheel axes and each figure wheel axis has a carry and warning axis in the same relative positions. This can be seen clearly from the plan views in 161, 164, 176 and 177. The repetition of this basic two–column addition unit in its unmodified form constitutes a fundamental desing flaw. This is described in detail below.

*Figure Wheels*

Each of the eight 31–digit numbers (seven differences and the tabular value) is represented by a column of 31 figure wheels with one figure wheel for each digit of the multi–digit number. The value of each digit is represented by the angular rotation of the associated figure wheel. Each figure wheel has forty teeth with the pitch of one tooth (9°) corresponding to a single decimal number 0–9. Each wheel therefore has four identical runs of decimal numbers and the same digit value can be represented by any one of four interchangeable angular positions of a wheel. The figure wheels rotate freely on the figure wheel shafts and rest on the figure wheel supports. These are comb–like pillars that run the full vertical length of the columns with the figure wheels resting on the equally spaced teeth of the comb. This is shown clearly on the extreme right top of 171 in the opened out view. In the lower view the supports are shown as rectangles with each column of figure wheels fitted with three figure wheel supports positioned roughly at 5–o'clock, 10–o'clock and 2–o'clock. The supports are not equally spaced around the figure wheel axis (see below).

The figure wheels are driven by drive arms keyed to the figure wheel drive shafts. The drive arms operate against internal nibs positioned at 90° intervals inside the barrel of each figure wheel. The figure wheels themselves remain on the same horizontal plane throughout i.e they do not move vertically but undergo circular motions within the cages formed by the combs of the figure wheel supports. The drive arms are engaged and disengaged by lifting and lowering the figure wheel axis at appropriate points in the cycle. The lifting action disengages the drive arms by raising them clear of the internal nibs. This occurs when the wheels are being added to and when manually setting the initial values at the start of a run of calculations.

As well as four internal nibs, each figure wheel has four external nibs integral with the barrel. The external nibs perform two separate functions: they run up against the zero stops (shown below the figure wheel supports shown at 2–o'clock) to ensure that the wheels do not overrun the zero position during giving off. They also set the carry warning mechanism by operating the curved warning limb of the carry lever if the figure wheel is driven beyond 9 during addition.

*Sectors*

Figure wheels in neighbouring figure wheel columns are coupled digit for digit by sector wheels assembled on axes that interpose between pairs of figure wheel axes. The sector wheels are free to rotate on the sector axes. They are driven by restoring arms which are keyed to the sector axes. The restoring arm rotates against a peg which protrudes above each wheel (see 171 upper right) and restores the sector to zero. The sector wheels and restoring arms are raised and lowered by the sector axes. Unlike the figure wheel drive arms which disengage from the internal nibs by being raised clear, the sector drive arms are fixed in the same plane as the peg and the drive arms and wheels are raised and lowered together with the sector axis.

The sectors can occupy three discrete vertical positions, fully disengaged, partially engaged and fully engaged. When fully raised the sectors are lifted clear of the figure wheels and disengage from both columns. In this position the two figure wheel columns are uncoupled from each other. This is the position shown in the opened out view in 171 top right. When half–lowered the sectors engage with the right hand column only and are disengaged from the left column. This partial engagement occurs because the teeth that engage with the right hand figure wheels are have double vertical depth. This is shown clearly on 171 top right. In the partially engaged position rotation of the sector is imparted to the figure wheel with which it meshes and this rotation is not transferred to the figure wheels to the left. Finally, when fully lowered the sectors mesh with both left and right hand figure wheels. In this fully engaged position rotation of the right hand figure wheels is transferred to the left hand figure wheels, wheel for wheel and tooth for tooth, via the intermediate sectors.

The rotation of the sectors is limited by the zero stops shown at the bottom of the view in 171. The zero stop performs two separate functions. It stops over–rotation of the sector by directly obstructing the path of the cut–away section of the wheel. It also provides a locking action when the sector is fully disengaged i.e. in the fully raised position when the sector is in the zero position a lug on the zero stop engages with a slot in the sector wheel. This locking action prevents derangement when the sector is not meshed with either figure wheel. The only detail of the sector locking slot is given in 171 where a small gap is indicated between the end of the stop attached to **M** and the root of the slot on the sector. The sector zero stop comb–pillars are not shown in the opened out elevation above.

*Locks*

The wedge–shaped parts alongside the figure wheel supports at about 9–o'clock are locks. These are sword–like devices which run the full length of the columns of figure wheels. The upper and lower ends of the lock are angled and pass through angled slots in the frame. The top end passes through an angled slot in the upper framing plate; the lower end passes through an angled slot in framing member below the odd figure wheel racks. The T–shaped framing member spans the length of the engine and is supported at each end by two front–to–back framing members at each end. The T–shaped section of this member is shown as **I** in 160 right centre, and in plan, with the angled slots for the locks, in 161 running right/left across the centre. Raising the lock withdraws the wedge from the between the teeth of the figure wheel. Lowering the lock drives the wedge into the inter–tooth gaps. During entry the wedging action corrects small rotational derangements of the wheel and when fully home locks the wheels in valid integral–value positions.

The locks operate on all figure wheels in any given column at the same time. If a wheel is deranged to the extent of presenting a tooth to the front of the wedge, the path of the lock is obstructed and the engine jams. These jams are intended to warn that a figure wheel is in an indeterminate position and thereby halt an operation that would lead to an incorrect result. The locks extend below the figure wheel columns and also lock the circular motion quadrants that impart the rotational motion to the figure wheel axes (171 lower right). (Locking the circular motion quadrants is shown in 160 left, centre). The locks are engaged and disengaged by the vertical motion drive in accordance with the timing cycle.

*Description of the Carry Mechanism*

The carry mechanism consists of the warning axis shown at 12–o'clock to the figure wheel axis, the three–limbed carry levers mounted on the warning axis, the spring–loaded warning mechanism with detent lever, the reset stops which are fixed to the figure wheel support shown at 2–o'clock to the figure wheel axis, and carry axis with the helical arrangement of carry arms.

The carry lever is the most complex single part in the engine and performs several different functions central to the operation of the calculating mechanism. Each carry lever has three limbs which project radially from a central boss with which it is forms a single piece. The carry levers do not pivot on the warning axis directly but are free to rotate on the boss attached to the detent support arm which is in turn keyed to the carry axis.

The carry lever can occupy four discrete positions the locations of which are determined by the position of four v–shaped detents in the detent limb of the lever (shown at 9–o'clock on 171). The spring–loaded detent lever which pivots on a spigot attached to the lever arm locates the lever in the four positions. These correspond to the four states, 'unwarned', 'warned', 'carried', and 'disconnected'.

Each figure wheel position has an associated carry lever keyed to the carry axis. The carry arms are arranged on a fixed vertical pitch equal to the pitch of the figure wheels and a fixed angular pitch of 22½° with a single 45° gap between the 15th and 16th carry arm only (see below for explanation). There are thirty carry arms in all, arranged in two runs of fifteen arms on a 22½° pitch separated by a double–pitch gap between the two runs. The helical arrangement of carry arms is shown at the top of the lower view in 171. In the plan view each arm shown hides its counterpart in the run below i.e. $\mathbf{C}_{30}$ conceals $\mathbf{C}_{15}$, $\mathbf{C}_{29}$ conceals $\mathbf{C}_{14}$ etc. The uppermost carry arm (at 4–o'clock) is annotated '$\mathbf{C}_{15}$ $\mathbf{C}_{30}$' indicating that the single arm drawn represents the upper and hidden arm below. For strict consistency each of the arms should have two notations though the first three have only one ($\mathbf{C}_1$, $\mathbf{C}_2$, $\mathbf{C}_3$). This is another respect in which the view in 171 is neither a true section view nor a plan but a diagrammatic hybrid. As the carry axis rotates, the carry arms sweep round and poll each figure wheel position in turn. Staggering the carry arms in the helical arrangement shown allows secondary carries to propagated i.e. a carry that results from a carry in the lower order decade can ripple upwards in what Babbage called 'successive carry' operation (see below).

The limb of the carry lever shown at 2–o'clock is the carry limb. The limb is drawn solid in the unwarned position and as dotted partial views in the other positions, warned, carried and disconnected (two positions). If the carry lever is in the unwarned position then the carry arm sweeps past without action. However, if the carry lever is warned then the locus of the lozenge at the end of the carry arm intersects with the warned position of the lozenge at the end of the carry limb so as to advance the carry lever to the 'carried' position. The motion imparted to the carry lever nudges the figure wheel in the next highest position by one tooth i.e. increments the next most significant digit by one. The figure wheel nudged by a lug on the third limb.

The third and remaining limb of the carry lever (shown at 6–o'clock) has attachments at two vertical levels. The two attachments can be seen in elevation in the opened out view at the top of 171, centre. The upper attachment has two lugs and resembles an escapement, the lower is a curved warning limb. The vertical position of the limb and the spacing between the two attachments is arranged so as to straddle two vertically adjacent figure wheels. In this way the escapement attachment and the warning limb of the same carry lever interact with different decades of the same number and carry action on a figure wheel occurs as a result of warning from the figure wheel immediately below: the lower attachment is operated by the outer nib of a figure wheel, and the upper attachment of the same carry lever operates on the figure wheel of the decade immediately above. The curved warning limb engages with the external nib of a figure wheel as the wheel passes from 9 to 0. The passage of the nib past the warning limb nudges the warning limb and advances the carry lever from unwarned to warned. This occurs while the figure wheel is being added to. During this phase of the cycle the escapement is raised clear of the teeth of the figure wheel immediately above and has no direct effect. After the addition phase the carry lever is lowered from its raised position for the carry phase. The warning limb can be still be operated by the external nib of the lower wheel with the carry lever in the lowered position. (This allows the carry lever to be advanced from unwarned to warned by secondary carries which may occur during the carry phase of the cycle, see below.) In the lowered position the lugs on the escapement are in the plane of the figure wheel teeth above and mesh with them in different ways to perform three separate functions.

1. With the carry lever in the *unwarned* the v–shaped cutout on the left hand lug engages with the figure wheel tooth as the carry lever is lowered and locks the figure wheel to prevent derangement during the carry cycle i.e. when the figure wheel is otherwise unsecured.

2. the carry lever is *warned* when lowered, the left hand lug is clear of figure wheel teeth but the right hand lug interposes between adjacent teeth and nudges the figure wheel one position on when the carry lever is advanced from warned to carried by the sweep of the carry arm. A carry occurs in the figure wheel as a result of a warning from the figure wheel immediately below. The right hand lug has an additional function: while the carry lever is in the warned position and waiting to be serviced by the rotation of the helical arrangement of carry arms the right hand lug acts as a partial lock.

3. with the carry lever in the *disconnected* position, the right hand lug again interposes between two adjacent teeth and acts as a figure wheel lock.

During the carry phase of the addition half–cycle the carry axis is lowered. If the carry lever is in the unwarned position then the cutaway on the left hand lug on the escapement engages with a tooth and locks the wheel. This ensures that the wheel does not derange when the locking wedges are withdrawn during the carry phase. If the position is warned then the right hand lug engages with the figure wheel and the motion imparted to the escapement by the rotation of the carry arm nudges the figure wheel one position on. The warning axis is raised during the addition half–cycle. In the raised position the warning limb can still be activated by the outer nib of the figure wheel but the escapement is lifted clear of the figure wheel above i.e. the lugs are disengaged leaving the figure wheel to free to be driven by the sector.

Motion can be imparted to the carry lever in four separate ways:

1. by the outer nib of the figure wheel as the wheel is driven past 9 during addition. This advances the lever from the unwarned to the warned position

2. by the sweep of the carry arm during the carry phase. This advances the lever from warned to carried

3. by the detent support arm. This returns the detent lever to the unwarned position by rotating the carry lever against the reset stops and driving the detent lever along the notched detent limb in the last detent.

4. by hand to the disconnected position. This last operation requires that the reset stops are raised to allow the lever to pass under the stop. The stops are then lowered to trap the lever. The stops are freed for raising and lowering by releasing the fixing screws which pass through slotted holes in the support bars for the stops. This is shown clearly in 171 top left. Disabling the carry mechanism in this way prevents carries from propagating past the disabled position. This has the effect of isolating the figure wheel column above the disabled position from the calculation below. Sections of the figure wheel columns can then be used for different purposes, as a cycle counter, for example (see below).

**The Calculating Cycle**

The operation of the calculating mechanism will be explained with reference to the layout of the basic addition unit shown in 171 lower left. At the start of the cycle the right hand figure wheel column will in general store a non–zero number. In the most general case each of the 31 figure wheels in a column will register a different angular displacement from the zero position. Similarly, the left hand column will in the general case have a non–zero number represented by appropriate initial settings of the figure wheels. The calculating half–cycle adds the 31–digit number from the right hand column to the 31–digit number on the left hand column and takes account of any carry. The transfer of numbers is non–destructive i.e. at the end of an addition half–cycle the number on the right hand column at the start–of–cycle is restored to that column, and the left hand column ends up with the sum of the two numbers.

*Initial Conditions*

The view shown in 171 lower left can be taken as showing the positions of the figure wheel drive arms and the sectors at the start of the cycle. Both figure wheel axes are rotated fully anticlockwise and the drive arms, in the raised position, are clear of the internal nibs. The figure wheel zero stops, which move with the figure wheel axes, are also clear of the external nibs. (The zero stop comb is shown n 171 top right (**S**). The coupling of the figure wheel zero stops and the figure wheel axes is shown in 160, left view, right. A yolk on the zero stop support engages with a colar on the figure wheel axis. This allows the axis transmit vertical motions to the zero stops but not circular motions.) The left and right hand figure wheels can be regarded as being shown set to zero. This is a special case and in not generally true i.e. in general the figure wheels will register a non–zero number at the start of the cycle and will be offset anticlockwise from the zero position shown, by a variable number of teeth (9° intervals) corresponding to the digit value for that number position. The internal nib shown at 5–o'clock will therefore in general be somewhere in the quadrant between 5–o'clock and 2–o'clock. This is true of both right and left figure wheels i.e. in general each of the 31 wheels in a column will be displaced anticlockwise from the position shown by a variable displacement corresponding the digit value. In the special case shown in 171 both figure wheels can be regarded as set at zero at the start of the cycle i.e. in the next addition half–cycle, zero (to 31 places) on the right hand column will be added to zero (to 31 places) on the left hand column. The figure wheel locks are full engaged as shown and both wheel columns are immobilised.

At the start of the half–cycle the sectors are against the zero stops (i.e. full clockwise), fully raised and with the restoring arms full anticlockwise. In the fully raised position the sectors are fully disengaged from the figure wheels. With the sectors in the home position (i.e. against the zero stops) raising the sectors locks them – this by engaging the locking lug on the zero stop with the slot in the sector. The sectors are against the zero stop at the start of cycle regardless of the numbers set on the right and left figure wheels at the start of cycle.

The warning axis is in the raised position as shown in the opened out elevation. In this position the warning limb of the carry lever can be acted on by the outer nibs of the figure wheel but the escapement lugs are clear of the teeth. The carry lever is in the unwarned position i.e. with the topmost detent engaged.

*Giving off and Addition*

The cycle commences with the lowering of the sectors into full engagement with the two figure wheel columns. Lowering the sectors unlocks them by disengaging the locking lug on the zero stop from the slot in the sector. In the same interval the right hand figure wheel axis and zero stops are lowered so that the drive arms are in the plane of the internal nibs and the zero stops are in the plane of the external nibs. The left hand figure wheel axis remains in the raised position with the drive arms and zero stops disengaged. This allows the external nib to pass the zero stop position which will occur if the number on the figure wheel exceeds 9.

The locks on both figure wheel columns are then raised i.e. disengaged, and the right hand figure wheel axis rotates clockwise from its rest position though its full travel of 81° i.e. 9° (one tooth pitch) short of a full quadrant. During this sweep the right hand wheel is reduced to zero and the zero stop prevents any overshoot by blocking the path of the outer nib. The number held of the figure wheel at the start of the cycle (in the case shown in 170 this is zero) is transferred tooth for tooth to the sector wheel which in turn drives the left hand figure wheel. The number on the left hand figure wheel is increased by the number on the right hand wheel and this is the basic operation of addition.

*Warning*

If during the addition the left hand figure wheel passes 9 then the external nib operates the curved warning limb and advances the carry lever to the warned position i.e. with the second detent engaged. The warning action is part of the giving off/addition phase and occurs only in digit positions that exceed 9 during addition and at different times within the addition window depending on the number value of the particular figure wheel. The process of giving off, addition and warning occurs simultaneously for all 31 digit positions in the columns.

The locks on both figure wheel axes then engage to correct minor derangements. At the same time the sectors are raised to their partially engaged positions i.e. they remain meshed with the right hand figure wheels and disengaged from the left hand figure wheels.

*Design Error*

In the description so far the right hand figure wheel rotates clockwise when giving off, the sector is driven anticlockwise and the left hand figure wheel clockwise during addition. These directions of rotation are inferred from the position of the sector restoring arm which is displaced anticlockwise to the drive peg. This indicates that the restoring arm provides active drive to restor the sector by clockwise rotation of the sector. It follows that rotation of the sector during giving off must be counter–clockwise from the zero stop as this is the only degree of rotational freedom possible. By this reasoning the right hand figure wheels rotates clockwise when giving off and, since it is meshed to the left hand figure wheel via the sectors, the left hand figure wheel therefore also rotates clockwise when being added to. This would indicate that the warning mechanism should be armed by clockwise rotation of the outer nib of the figure wheel during addition. However, with the warning mechanism shown in 171 the curved warning limb will be correctly operated by *anticlockwise* rotation of the figure wheel not clockwise rotation. If the figure wheel were to rotate clockwise with the arrangement as drawn, the curved warning limb would foul the outer nib of the figure wheel and act as a stop. Clockwise rotation of the figure wheel during addition is therefore inconsistent with correct warning action which requires advancing the carry lever from unwarned to warned and the mechanism well not work if made as drawn.

The directional arrows drawn on the five axes in 170 are little help as these are inconsistent however interpreted. So as to use 171 to complete the description of the calculating cycle it will be assumed that the left hand column rotates anticlockwise during addition for the correct operation of the warning mechanism. The measures taken to remedy this apparently serious design flaw and alternative interpretations of the layout in 171 will be explored in the section describing the modern implementation.

*Restoration*

The end of the giving off and addition phase leaves the left hand figure wheel with the sum (without carry) of the two initial values of the two figure wheels and the warning mechanisms armed in the digit positions in which the figure wheels exceeded 9 during addition. The right hand figure wheels are at zero with the drive arms rotated fully clockwise. The right hand figure wheels have therefore lost their initial values. However, the sectors are displaced from zero by the number of teeth equal to the number value on the right hand figure wheels at the start of the half cycle and the values are restored to the right hand figure wheels in the next phase of the half–cycle.

The locks disengage from the right hand figure wheels and the sector restoring arms rotate in a clockwise sweep. During the sweep the restoring arms engage with the pegs on the sectors and restore the sectors to zero. The sector zero stops prevent any overshoot by obstructing the cutaway section of the sector. Since the sectors are disengaged from the left hand figure wheels these remain unaffected by the restoration of the sectors. The right hand figure wheels are driven by the sectors and the restoration of the sectors to zero restores the figure wheels to their positions at the start of the cycle before giving off. During the restoration process the right hand figure wheel axis rotates anticlockwise to the position shown in 171 i.e. returns the drive arms to the position at the start of cycle. This does not affect restoration as the figure wheel axis is in the raised position i.e. the drive arms are lifted clear of the internal nibs. The figure wheel locks then engage to correct minor displacements and prevent spurious derangement.

*Carry*

During the carry phase the warning mechanisms are serviced in turn and the requisite carries propagated upwards i.e. the next decade up is conditionally incremented by one depending on whether the associated warning mechanism is armed or not.

The carry phase commences with the withdrawal of the locks from the left hand figure wheels which occurs immediately after the addition phase in the first half–cycle. At the same time the warning axis is lowered. This lowers the carry limbs into the same plane as the carry arms (the lozenges of the warned limbs are now in the path of the lozenges of the corresponding carry arms), and the escapement limbs are lowered into the same plane as the figure wheel teeth. If the position is unwarned then the left hand escapement lug is active i.e. the v–shaped cutaway locks the figure wheel during the period in which it is otherwise unsecured. If the position is warned then the left hand lug is clear of the teeth and the right hand lug is active i.e. lowered so as to partially interpose between two adjacent figure wheel teeth. This acts as partial lock while the position awaits its turn in the successive carry sequence.

The rotation of the carry axis follows and with it the rotation of the helical array of carry arms. The direction of rotation is anticlockwise as indicated by the direction arrow drawn in the circle representing the carry shaft. Each carry limb is polled in sequence by the corresponding carry arm starting from the below and working upwards. If a carry lever is unwarned then the trajectory of the carry arm lozenge does not intersect with the position of the carry limb lozenge. The lozenges pass without contact and no action to carry is taken. However, if the position is warned, the locus of the two lozenges intersect. The outer face of the carry arm lozenge wipes past the inner face of the carry limb lozenge and pushes the carry limb aside as it slides past. The action of the carry arm nudges the carry limb one position on i.e. into the carried position which is the next discrete position marked by a detent. The clockwise rotation of the carry lever advances the right hand lug of the escapement and this nudges the figure wheel one tooth pitch on i.e. increments the number value by one. The helical arrangement of carry arms services each warning mechanism in turn and increments the figure wheel depending on whether or not the mechanism is warned. The net result of this conditional action is that mechanisms warned by figure wheels during addition increment by one the figure wheels of the next higher decade as required for correct multi–digit addition with carry.

With the carry lever in the disconnected position the right hand lug interposes between two adjacent figure wheel teeth when lowered and acts as a figure wheel lock during the carry phase.

Both sets of carry axes (i.e. odd and even axes) rotate together. During the even carry the odd carry levers rotate freely without encountering any warning limbs and vice versa. This is because during the carry phase of one set of axes the warning axes of the other set are raised and the warning limbs and swirling carry arms are in different planes and do not interact.

*Secondary Carries*

A primary carry is one that occurs as a result of a warning mechanism being armed during addition as described above. A secondary carry occurs when a figure wheel exceeds 9 as a result of primary carry. During a primary carry phase the figure wheel at 9 will be nudged on to zero by the right hand lug of the escapement as its position is polled by the sweep of the rotating carry arm. As the figure wheel passes 9 the outer nib of the figure wheel passes the curved warning limb, and arms the next higher warning mechanism as before. The newly armed warning mechanism is immediately polled by the next carry lever in the sequence and the carry is propagated as before. Ripple carries can be propagated up the stack in this way. The worst case ripple condition occurs in a column of figure wheels all set to 9 and a warning mechanism armed in the least significant position. The successive carry operation described will correctly propagate the carry up the stack leaving each figure wheel set to zero and an overflow carry at the top of the stack.

There is a feature of the geometry of the arrangement of carry arms that has yet to be explained. In the helical arrangement of carry arms the vertical separation of the carry arms corresponds to the pitch of the figure wheels and the angular pitch is fixed at 22½°. There is one exception to this i.e. the spacing between the 15th and 16th arms (counted from below) is 45° not 22½° (gap shown at 5– o'clock in 171, between plan of $\mathbf{C}_1$ and $\mathbf{C}_{30}$. The function of the gap is to prevent the carry arm and carry limb from fouling in one of two conditions. During giving off the warning axis raised as shown in the elevation 171 top left.
In this position the lozenges at the end of the carry arms are not in the same plane as the lozenges at the end of the carry limbs and there is no danger of the carry limb and arm if a carry lever advances from the unwarned to warned positions. However, during the carry cycle the warning axis is lowered. If the carry arms were on a fixed 22½° pitch then the lozenge of the 16th carry limb, if warned, would foul the lozenge of the corresponding carry arm when the axis lowered. In addition to axial fouling of this kind there is the danger of rotational fouling in the case of secondary carries. If the 15th carry limb was unwarned at the start of the carry cycle but received a secondary carry then the lozenges would foul as the 16th carry limb advanced to the warned state. This was foreseen in the original design and Babbage allowed a double–pitch gap between the 15th and 16th carry arms to allow sufficient clearance to avoid axial or radial fouling.

*Resetting*

The end of the carry phase coincides with the end of the restoration of the right hand figure wheels to their initial values at the start of the half–cycle. Both left and right hand figure wheel locks then engage and the sector axis is raised into full disengagement from both left and right figure wheels. The end of the first half cycle leaves the right hand figure wheels with their start–of–cycle values restored, the left hand figure wheels with the sum of the left and right hand initial values, and the sectors against their zero stops, fully disengaged and locked.

There are two last actions needed to restore the mechanism shown in 171 to the start–of–cycle condition in preparation for the next repetition of the first half cycle: returning the sector restoring arms to their anticlockwise home positions; and resetting the armed warning mechanisms to their unwarned positions. Both these actions occur during the second half–cycle. The sector restoring arm is returned to the home position at the start of the seconcd half–cycle during giving off. Resetting the left–hand warning mechanisms to the unwarned positions occurs at the end of the second half cycle while the warning axis is raised and the escapement is out of the plane of the figure wheel teeth. The warning axis rotates clockwise and with it the detent support arms which are keyed to the axis. The spring–loaded detent lever, mounted on the detent support arm, is carried clockwise and the carry limbs are driven towards the reset stops by the detent lever. The carry limbs in the carried position will be driven against the reset stops and their motion halted. The detent support arm continues clockwise and the detent lever is driven in turn from engagement with the carried detent to the warned detent and finally to the unwarned detent as it rides along the notched outer curve of the detent limb. The travel of the detent support arm is such that the carry levers already in the unwarned position at the start of the reset process do not bear on the reset stops and remain unaffected. The clockwise rotation of the detent support arm is followed immediately by a counter clockwise return motion which restores the unwarned carry levers to their home positions as shown in 171. The carry levers in the disconnected position are not reset to the unwarned state: these are trapped behind the reset stops and simply wave back and forth during the movement of the detent support arm. The extremities of travel of the disconnected carry limbs are shown by the two dotted positions bracketed opposite the *'Disconnected'* label. The clockwise/anticlockwise waving motion of the warning axis resets the carry levers to the unwarned position. This is the only function of the circular motion of this axis and Babbage refers to the axis as the *'Unwarning Axis'* in the timing diagram BAB [F] 385 1a.

*The Two Half–cycles*

During the second half cycle the left hand figure wheels give off to the column to the immediate left (not shown in 171) in a repetition of the sequence described. As noted earlier, the progression from column 1 (or the lowest numbered active column) through to the tabular value on column 8 does not occur in a stepwise sequence right to left from column to column in seven distinct repetitions of the same sequence. Rather, all the odd–numbered columns are siulutaneously added to the even–numbered columns in the first half cycle and all the even–numbered columns are added to the odd–numbered columns in the second half cycle. The phasing of the odd–to–even and even–to–odd half–cycles and the interleaving of the various actions within a cycle are shown in the timing diagram BAB [F] 385/1a. This diagram is fraught with inconsistencies and errors but will serve for the meanwhile as a general guide to the timing of the events in the two half cycles.

## Circular Motions

### General

The calculating cycle requires intermittent circular motions of the figure wheel axes, the sector axes, the warning and carry axes. The circular motions are smooth (i.e. have no sudden discontinuities), intermittent and reciprocating. The circular motions are executed in turn by the odd and even axes in a phased sequence as specified in the timing diagram. The circular motions of the carry axes which drive the helical arrangement of carry arms are not derived from the circular motion cams but from an intermittent drive mechanism coupled to the main drive socket. This is described in the Drive section; this section describes the circular motions of the figure wheel, sector and warning axes.

Figure wheel circular motions occur during giving–off and during restoration by the sectors of the number given off. During giving off the figure wheel axes rotate the figure wheels to the zero stops by driving the giving–off arms against the internal lugs of the figure wheels (171). After giving off, the vertical motion mechanism raises the figure wheel axis. This disengages the arms from the internal lugs by lifting them clear. In the raised position the figure wheel axes return the giving–off arms to the home position until lowered to reengage at the start of the next calculating cycle. The circular motion therefore consists of two separate sweeps, one a drive stroke and one a return stroke, separated by timing gaps and phased with the vertical motions.

The sectors store the number given off and, after a timing interval, restore it to the figure wheels. During giving off the sectors are driven by the figure wheels. When restoring, the sectors are driven by restoring arms which are keyed to the axis and drive against the internal lugs of the sectors. The sector axes and the sectors are lifted clear of the lugs by the vertical motion mechanism and the restoring arms return to the home position ready to be lowered for the next cycle. The circular motion of the sector axes therefor consist of a drive stroke which restores the number to the figure wheel and a return stroke, separated again by timing gaps and phased with the vertical motions.

The circular motions of the warning axes reset the carry warning mechanisms by a forward sweep followed immediately by a return sweep which restores the mechanism to the unwarned position.

The circular motions are derived from the six pairs of conjugate cams located on the upper section of the cam stack (163). The twelve cams drive cam followers which rock on pivot shafts (159). The arrangement converts the continuous circular motion of the cams to reciprocating motion of the follower arms. The six sets of follower arms drive links to six long backing bars each of which carries four toothed racks which slide in straight machined channels. The backing bars join the racks but do not themselves slide in guide channels. The original design calls for six sets of four in–line racks, one set for each of the four odd figure, odd sector and odd warning axes, and set each for the three sets of even axes (see 'Modern Implementation for modification of racks for the warning axes). The arrangement translates the reciprocating circular locus of the follower arm extensions into linear reciprocating motion of the racks. The racks mesh with sector gears which convert the linear motions back into the required intermittent reciprocating circular motions for the axes (171, lower right).

### Description

*Drawing References*

Main drawings are 159 left and 170, also 160 and 163 for the vertical cam layout; 161 extreme right shows a partial detail of a sector wheel circular motion follower arm; 169 left is a printing cam and its significance is unknown. Modern drawings: the B–series.

The general layout is given in plan in 159 left which shows a set of three follower arms and partial cam profiles. The pivot for the sector follower is shown at ten–to–twelve, the warning follower pivot at five past twelve, and the figure wheel follower pivot at twenty–five to. Except for a rotational offset of the cams the layout is identical for odd and even motions and 159 shows only one set cams and followers i.e. the arrangement is duplicated for the alternate set but is not shown and the followers for the unshown use the same pivot shafts. 170 is a clearer version showing the full cam profiles for the even difference cam pair and follower in one view, with the arrangement for the other two motions shown separately on the left. The view on the right corresponds to the plan view of the uppermost two cams in the stack.

It is not immediately clear whether the layout shown on 159 left is intended to represent the odd or even cams or whether this view is intended to serve for both. One way of resolving this would be to establish the top–to–bottom sequence of cams. However, the convention of using dotted lines to indicate hidden edges is not strictly adhered to and differentiating the planes of the followers to establish how they are layered is not obvious. The sector and figure wheel followers in 159 (left), for example, are in different planes but are both drawn with unbroken lines and superimposed; the same is true for the two rollers at five–to–twelve. There are other signs of draughting inconsistencies: the two arms of the same figure wheel followers are shown in the same plane in 159 lower left, and in different planes in 170.

Unlike the vertical motion cam followers which have sliding contacts (sector followers are an exception, see Vertical Motion section) the circular motion followers use rollers throughout. In the case of the sector and figure wheel followers the links are driven by a simple extension of the follower arms. In the case of the warning followers the links are driven by an additional arm, integral with the follower arms. Unlike the vertical motion followers, where the arms for a cam pair and the associated bar lever are vertically staggered on the pivot shaft, the arms for the circular motion followers and the drive to the link are all in the same horizontal plane. For both types of circular motion followers the two rollers of a follower set are pivoted on either side of the arm so as to span the distance separating the two cams (160 shows this though identifying parts of the same assembly from this crowded drawing takes some hard peering). The drive extension arms have slotted forked ends which take pivots for the links to the rack assemblies.

The three lowermost cam pairs of the upper twelve cams in the stack generate the odd axes motions in the order bottom to top, sector axes, figure wheel axes, and warning axes. The uppermost three cam pairs generate the even motions in the order top to bottom, sector, figure and warning axes. So the upper and lowermost circular motion cam pairs generate the even and odd sector motions respectively, and the two cam pairs generating the odd and even warning axes motions are adjacent to each other in the centre.

The spacing between two cams of a conjugate pair of circular motion cams is wider than for the vertical motion cams. The figure and warning cams are a standard 1.25" apart; the separation of the sector cams is 0.75" (160, 163). So the two outermost pairs of circular motion cams (odd and even sector cams) are closely spaced and these sandwich the four more widely spaced cams between them. The wider spacing of the figure and warning axes cams allows the forked ends of the extension arms to pass between the cams. This is shown clearly on 170 left where the follower arm is shown on the inside of the roller. In the case of the sector arms, the forked ends are outside the largest cam diameter which allows the smaller spacing (170 right). The rollers on the figure and warning follower arms have longer bosses to span the separation of the of the more widely spaced figure and warning cams (160).

*Description of Motions*

Each figure wheel has four decades of numeral 0–9 i.e. 40 teeth with each decade occupying one 90° quadrant of a full circle. The maximum rotation of a figure wheel is 81° i.e. 9 increments of 9° representing a ten–digit interval (171). The internal lug occupies the remaining 9° sector. The circular motion of the figure wheel giving–off arms is therefore a fixed sweep from the home position to the zero stop position during giving off. This is followed by a vertical lift to disengage from the internal lug, and return to home while the number given off is restored by the sectors. The width of the internal lugs is kept sufficiently small to prevent the giving off arm interfering with the lugs when the arm is next lowered after the return–to–home stroke.

The sector restoring arms execute a similar pattern of circular sweeps. In the fully lowered position the sector wheels engage the two adjacent figure wheel columns and giving–off right to left takes place. The maximum angular displacement of a sector wheel occurs if a '9' is given off. In this case the sector is moved through 81° from the zero stop position. However, the rest position of the restoring arm is shown 84° from the zero stop position (171). The 3° margin prevents the lug and the arm interfering by ensuring that the arm is well clear of the worst–case angular displacement of the internal lug. The '81°' inscribed in the circle representing the sector axis in 171 refers to the maximum displacement of a sector wheel and not the total angular sweep of the restoring arm.

This 3° clearance for the restoring arm accounts for the slight difference in length of the sector and figure wheel follower arms on 159 though the cam throws are identical. The distance from the pivot to the forked end of the figure wheel follower arm is 9.0". This is extended to 9.3125" for the sector follower arm. This gives a 0.1" longer stroke to the sector rack which corresponds to about 3° safety clearance for the sector restoring home position.

*Implementation*

The numbering convention adopted for the vertical motion cams was extended to the circular motion cams. The vertical motion cams are numbered 1 through to 16 starting from the bottom of the stack. The circular motion cams continue the series from 17 through to 28, this last being the topmost cam of the stack. The circular motion cams and their functions are as follows:

28 even sector arm restore
27 even sector arm return
26 even figure wheel arm give off
25 even figure wheel arm return
24 even warning detent restore
23 even warning axis return
22 odd warning axis return
21 odd warning detent restore
20 odd figure wheel arm return
19 odd figure wheel arm give off
18 odd sector arm return
17 odd sector arm restore

*Modifications*

The omission of the carry and warning axes for the 7th difference column removes the need for a rack for the warning axis circular motions. The rack for the first odd warning axis was omitted and the backing bar for the odd warning axes therefore has three racks instead of the four originally specified.

Because of the introduction of mirroring of alternate columns the direction of the active strokes and return strokes are reversed for the odd and even axes. For example, even sector restore stroke (170 left) occurs when the link pivot drives from right to left. In the duplicated arrangement for the odd sector axes (not shown) the odd sector restore stroke is from left to right.

*Cam profiles*

The cam profiles were generated from the known requirements for the circular motions. The overall shapes were confirmed on 159 and 170 by visual inspection rather than exact tracing and matching. The timing windows for the circular motions of the figure wheel, sector and warning axes were all shortened slightly to accommodate the locking action (see Vertical Motion section) and the detail of the original shapes was therefore in any event subject to minor alteration.

The even sector cams (Cams 28 and 27) illustrate the general relationship between cam shape and cycle timing. 170 right shows the position of the cams at the end of the even sector axes return stroke i.e. with the link at the right hand extremity of travel and the roller of Cam 27 at the top of the rise. This corresponds to 66° into the cycle (337B31). The even sectors do not rotate for the next 294° as indicated by the long sections of constant radius on both even sector cams. During this period the sectors undergo various vertical motions: they are lowered from their fully raised positions into full engagement for giving off odd–to–even and then lifted into half–engagement ready for the restore stroke. Cam 28 then becomes the active cam and drives the sectors to the zero stops so completing the stroke which restores the number given off to the figure wheels. The end of the stroke occurs 354° into the cycle (timing diagram and 337B31). In the case of the circular motion cams the timing diagram and the angular distribution of events round the cam correspond exactly; in the case of the vertical motion cams, there were occasionally slight deviations. A short dwell of 12° follows during which the sector axis is fully raised. This corresponds to the sections of the cams where the profiles coincide (170 right view, bottom centre). The dwell occupies the interval 354° to 6°. The return stroke follows which occupies the interval 6° to 66°. During this period Cam 27 is active.

## Vertical Motions

### General

The calculating cycle requires intermittent vertical motions for the figure wheel axes, sector wheel axes, locks and warning axes. These vertical motions are derived from the eight pairs of conjugate cams located in the lower section of the cam stack. Cam followers operate three pairs of horizontal bars slung on the underside of the engine and one pair midway up the cam stack. The arrangement of cams and cam followers converts the continuous rotational motion of the cams into intermittent reciprocating motion of the horizontal bars. The reciprocating motion of the horizontal bars is converted to vertical motion by bell crank levers (163, detail on 160). These lift and lower the axes and locks. The overall arrangement produces four sets of intermittent vertical motions phased according to the timing diagram.

The 'vertical motion' section covers the vertical drive for the locks, figure wheel axes, sector wheel axes, warning axes, as well as the cams, horizontal bars, bell crank levers, and additional framing support members.

### Description

Drawing References:

Main drawings are 159, 160 and 177. Also 161, 163, 167, 168, and 169.
E–series.

*Action of the Cams and Followers*

The vertical motion cams are the eight pairs of closely spaced cams in the lower section of the cam stack (160 and 163). Eight pairs of cam followers rock on eight separate pivots and are driven by the irregular shapes of the cams (159 shows all 16 cam followers and the pivot layout; 168 and 169 show clearer partial views). Bar levers, which are integral with the cam follower bosses, slot into the horizontal bars. Six of the follower pivots project through the lower framing plate of the cam stack to enable the bar levers to engage with the drive slots of the horizontal bars. The arrangement of cams, cam followers and bar levers, convert the continuous rotational motion of the cams into intermittent reciprocating motion of the bars.

The levers operating each of the eight horizontal bars is driven by two cam followers on the same boss. Each of the cam follower arms is driven by a separate cam. Cams therefore occurin conjugate or complementary pairs with one pair of cams associated with each of the eight vertical motions. The purpose of the complementary cam arrangement is to provide positive bidirectional drive for the vertical motions. The motion of a cam follower boss is determined by the outside surface of the active cam of the pair and the rises acting on the leading and trailing arms dictate these motions. The leading arm provides drive in one direction and the trailing arm, driven by the mating cam, provides the return motion. The naming convention adopted is that the leading arms point in the clockwise direction as viewed from above (the cams rotate anticlockwise so viewed). The leading arm does not necessarily make contact with the active cam first. At any given time only one of the pair of cam followers is in direct contact with the cam. The other follower is provided with a clearance from the cam surface.

The overall layout of the vertical motion cams and followers is shown in plan on 159 with additional partial views on 168 (left) and (right). 159 gives partial cam profiles with all significant rises shown in local relation to active portion of the cam follower. The eight follower pivots are shown distributed around the cam stack on a fixed diameter pitch circle. The cams are shown with standard outside diameters. The shape of the cams deviates inwards from the standard outside diameter.

In six of the eight cases the contact between the cam follower and the cam circumference is shown as a sliding contact (159). Only in the remaining two cases are the cam followers shown with a roller on one of the two arms and a sliding follower as the other. Roller–cum–slider followers are provided for odd and even sector motions only. This is shown on 159; the spaghetti junction of intersecting followers with rollers is clarified in 168. In all cases (i.e. sliders and rollers) leading arm rises are shown as straight and trailing arm rises as are shown curved on a 2.5" radius. The straight plane surfaces of the leading sliding arms are shown near–parallel to the rises on the cams (see for example the even warning follower on 159 top); in the case of the trailing arm rises, the shape of the inside of the arm roughly matches the curve of the rise (see even figure wheel follower on 159 lower right). It seems that Babbage was concerned to avoid point loading by distributing the load over a contact surface. The trailing arm rises are radiused to avoid fouling: if the trailing rise was straight, the top corner of the rise would foul the inner surface of the arm with the worst case occurring about half way through the rise.

In all cases of sliding followers the distance from the centre of the cam follower pivot to the line of contact on the sliding arm is shown as 3.50". In the case of the roller followers the 3.5" is taken from the pivot centre to the centre of the roller rather than to the line of contact. In the case of sliding contacts, the line through the pivot centre and the point of contact is in each case tangential to the outside diameter of the cam i.e. with the follower at the top of a rise. The effective angle between the follower arms on the same boss is therefore constant. In the case of the two roller–cum–slider followers the line through the point of contact of the roller and the pivot centre is tangential. Here the effective angle between the follower arms differs from that of arms with sliders only.

It is not immediately obvious why rollers were preferred for the sector vertical motions. The sector wheels represent the largest deadweight to be lifted, while the locks, though lighter, represent the largest shock load to the drive. It is possible that Babbage sought to reduce wear by using a roller on what he considered to be the most demanding load. This view is supported by the fact that in the case of the circular motions, rollers are preferred on all the cam followers, and sliding contact is avoided entirely. Though the loads for the circular motions are lighter, the duration of the actions tend to be more sustained. The rises on the cams are therefore longer and the period of active sliding contact, and therefore of wear, is correspondingly greater. It is possible that Babbage preferred rollers to sliders wherever possible. However, rollers are less space efficient: they require greater separation between the cams than do sliders and a lower stacking density of the cams would be required to accommodate them. The compressed space of the lower cam stack assembly as drawn prohibits the general use of rollers. However, the tight pitch of the cam spacing is relaxed in the case of the sector cam followers because they require a free vertical motion to disengage during setting up. It is likely that Babbage exploited this additional space to provide rollers where he could i.e. one for each of the sector followers. An alternative to the general use of sliders would have been to reduce the overall stacking density of the vertical motion cams and expand the lower section of the cam stack. Without an additional stage of linking this would increase the overall height of the machine and the knock–on effect would be to put the upper figure wheels out of normal reach. It seems that Babbage preferred sliders to an overall increase in cam stack height.

*Lifting and Lowering*

The lifting and lowering action for the figure wheel axes, warning axes, and the locks is provided by bell crank levers which are slotted into the horizontal bars below. The bell cranks for the sector axes are operated from above (160)(163 shows the even sector bar but not the bell cranks).

The bell cranks for the figure wheel axes, warning axis and sector axes have forked ends which fit into bobbins pinned to the lower end of the axes (160). The fork and bobbin arrangement allows independent vertical and circular motion to be imparted to the axes without conflict: the axes are lifted by the forked crank in the bobbin and rotated in the fork by the racked drive (see section on Circular Motion). The basic action of the bell crank and horizontal bar is clearly shown for the figure wheel axis bearing the tabular result (extreme left hand axis on 163 with detail on 160). The lower end of the axis projects below the bobbin through a bearing hole in a plate fixed across the underside of the front and rear framing members (160 lower left). The arrangement for the locks is slightly different. Unlike the axes, the locks do not rotate so a bobbin drive is not called for. Instead the lifting and lowering action is transmitted by a connecting rod pivoted at each end to the lock and at the other end to the bell crank (160). The connection between the sector bars and the sector bell cranks is shown as a mix of pivots and slots. 163 shows the even sector bar with pivots for the first and last bell cranks, and with slots for the two intermediate cranks. 160 confirms pivots for the last odd and last even crank drives.

The layout of the bell cranks is shown on 177. The bell cranks for a figure wheel axis, sector axis, and warning axis share a single shaft for each of the eight column positions (160 and 177). The lock cranks each have a separate shaft. The bell crank shafts are located at each end in bearing holes in the horizontal framing members. The longer of the forked bell crank levers drive the sector axes; the shorter ones drive the figure wheel axes (middle) and warning axes (rear) (177).

The figure wheels are set to their initial values manually. To free the figure wheels for setting up they must be disengaged from the sector wheels. Two lifting handles are provided for this purpose. The arrangement is shown on 163 and 177. Lifting the left hand handle operates shaft to which the bell crank lever for the first odd sectors is pinned
(pinning shown on 161). This bell crank lever drives the odd sector bar to lift the remaining odd sector axes. The right hand handle lifts the even sector axes in the same way.

As well as disengaging the sector wheels from the figure wheels when setting up, it is also necessary to disengage the sector bars from the horizontal drive. This prevents the drive for the vertical motions conflicting with the now immobilised sector axes when the engine is turned over during the setting up cycle. Details of the mechanism for disengagement are given on 159 centre, and 168 top right (163 shows only the support bracket on the cam stack side of the right upright). The sleeve, cam follower and bar lever for each of the odd and even sector bars are an integral assembly which is free to slide upwards on the cam follower pivot. Operating the release lever raises the two sleeves and lifts the bar levers out of their drive slots. Two detents for the release lever spring hold the lever in the released and unreleased positions.

**Implementation**

*Support for Horizontal Bars*

The six lower horizontal bars, which provide the vertical motions for the figure wheel axes, warning axes and locks, span the full length of the frame. The bars are supported at each end by slotted supports fixed front to rear to the main frame uprights. Five of the bars pass under the cam stack and are and these have an additional support on the underside of the right hand side of the cam stack. This slotted support, fixed to the right hand print shaft bearing mounting, is shown in 167, 163 and on 159 (faint dotted). The five longer bars are associated with the vertical motions for the odd warning, odd and even figure wheel axes, and the odd and even locks (159).

There is very little separation between the horizontal bars for the even figure wheel motions and the odd lock motions (159) and 167 shows the bar support with a thin tongue separating the two bars. This was considered a weakness and, given the material (phosphor bronze), difficult to manufacture. The thin tongue was omitted in all four bar supports and single broader slots provided for the two bars. Separation was instead provided by five spigotted phosphor bronze spacers. These are located in equally pitched holes along the length of the odd lock bar. The original separation remains unchanged. The spacers are 5/8" diameter and 0.082" thick (E323).

During assembly it was found that the full 62" span between the main uprights was too long to avoid downward bowing of the horizontal bars. An additional intermediate support was added for the six bars at the approximate midpoint of the span between the main uprights. The extra support is identical to the two fixed across the uprights.

*Counterbalancing the Locks*

During assembly it was found that even with a four–to–one reduction in the drive the engine could not be turned past the points in the cycle where the locks are released. The weight of the locks, friction in the angled slot–bearings, sideways pressure from the figure wheels, and the suddenness of the motion, make releasing the locks the most demanding load in the cycle. The 45° pressure angle on the lock cam presents a shock load to the drive which was too great to overcome with the original arrangement.

The solution adopted was to counterbalance the weight of the locks using springs. The original design (163) shows a single figure wheel axis projecting above the upper bearing plates and passing through a spring to counterbalance its weight. This single feature was assumed to be generic and the technique was used to counterbalance all figure wheel, sector wheel and warning axes. However, counterbalancing the locks from above in the same way poses difficulties. The locks are not shown projecting above the upper bearing plates and the shock load problem was encountered after manufacture i.e. during the build. Extending the locks would have meant remaking the locks and modifying the bearing plates. An additional difficulty is presented by the fact that the motion of the locks has a sideways as well as vertical component: the locks are lifted by the bell crank levers and slide in angled bearing slots which give a sideways motion so as to withdraw from the figure wheels (160, 161). In addition, the locking mechanism for the first column (seventh difference, extreme right) was modified to immobilise the first column during a part of the cycle in which it is unsecured (see below, pg. 6). The correct operation of this mechanism relies on the weight of the lock. If each lock was counterbalanced individually, this additional lock would need to be excepted.

These difficulties weighed against counterbalancing the locks using individual springs acting on the upper bearing plates. Instead, a single spring–loaded assembly was devised to act on the horizontal drive bars to counterbalance the locks via the bell cranks. This relatively small assembly is visually discrete and for the most part passes unnoticed.

The counterbalancing mechanism consists of a set of three springs placed end to end. The springs are in compression and kept in axial alignment by a forked tie rod (E411) threaded along its length and a threaded sleeve (E419) passing through the centre of the springs. The tie rod acts on a lever which is trapped in a recessed block (E414) screwed to the side of the horizontal lock bar. The effect is to bias the bar to the left i.e. in the direction of lift.

The compression force is adjusted by turning the threaded sleeve and shortening or lengthening the effective length of the composite spring. There are two identical mechanisms housed in the one assembly – one for the odd axis locks and one for the even axis locks. The six springs are of the same type used on the upper bearing plates to the counterbalance the figure wheel, sector wheel and warning axes.

*Modification to the Sector Bars*

The bar driving the sector bell cranks is shown supported only by the bell crank levers (160 and 163). The cranks for the first and last even axes have pivot fixings and the two intermediate cranks are shown with slot drives. The arrangement is assumed to be repeated for the odd sector axes (in 163 the odd sector bar is masked by the even sector bar). As drawn, the sector bars would ride in a slight arc determined by the length of the bell crank arm, and the intention seems to be that the intermediate cranks would lose the vertical component of motion in the slots. This arrangement gave rise to two concerns. The effective length of the crank with the slotted drives is shorter than that of the pivoted cranks, and the vertical motions imparted to the first and last sector axes would therefore differ slightly from those imparted to the intermediate sector axes. Deepening the slots to equalise the lever lengths would weaken the bar to an unacceptable extent and would risk fouling the slots at the extremities of the travel. A further concern was the risk of vertical bowing due to the upthrust from the bell cranks.

The sector bars and associated bell cranks were modified to take account of these concerns. The extreme left hand pivot connection between the bell crank and the bar was retained as shown in the original. All other slot drives, including the two closest to the cam stack (odd and even sector bars), were replaced with pivots and slightly elongated holes in the bars. With this arrangement, the vertical component of motion imparted by the arc of the bell crank lever tends to diminish over the length of the bar. Elongating the holes in the bars avoids any contention that might arise from small differences in the lengths of the levers, and the use of pivots in preference to slots avoids the risk of disengagement from upward bowing.

An additional mechanism was provided to control the operation of the lock for the first odd figure wheel axis (see below 'Modified Seventh Difference Lock', pg. 8). A further advantage of replacing the right hand fixed pivots with elongated pivots is that the downward motion imparted to the even sector bar by the additional first odd lock mechanism (E22) does not conflict with the upward arc that would otherwise be imparted by the bell crank.

An additional slotted support was fixed to the right hand uprights to provide front–to–rear support for the sector bar close to the cam follower. The purpose of this modification was to spare the bell cranks taking any side thrust from the cam follower lever acting in the profiled slot at the end of the sector bar, and also to reduce the risk of disengagement during normal operation.

*Modification to the Drive for the Warning Axis*

The modification of the axes layout for mirroring [see Calculation section] only affects the position of the carry axes so the layout of 177 is unaffected with the exception of the bell crank for the first odd warning axis (seventh difference, top right on 177) which was omitted. The drive slot for the horizontal bar for the axis was also omitted.

*Cross–over Bar Levers*

There are three instances in which the bar levers which drive horizontal bars need to cross over bars in the same plane. These are the bar levers for the odd locks, even warnings and even figure wheels (159). The bars for these three motions have raised projections with slots for the bar levers as shown for the even sector bar on 163. Raising the drive slots allows the bar levers to clear the intervening bars and provide drive to the otherwise obstructed bars.

The lower cam stack framing plate is drawn ½" thick (160) with just sufficient clearance for the vertical projections on the three bars. The annotation on 160 indicates that the framing plate should be thickened from ½" as drawn, to 1". If the upper surface of the framing plate is taken as the reference and the plate thickened downwards, there would be insufficient space for the slotted projections on the horizontal bars (E324, E342, E334). This was resolved by following the vertical layout arrangement on 163 which is drawn with the plate thickened and with sufficient clearance for the bars.

*Disengaging the Sectors for Setting Up*

Babbage's annotated instructions for setting up BAB[F]385 suggest that the sectors should be 'lifted to their highest point' during the setting up procedure i.e. the sectors should be taken out of engagement. The sectors are disengaged during setting up by operating the two sector lifting handles (163, 177). No provision is made on the original drawings for locking these in place to hold the sector axes in their raised (disengaged) positions so as to relieve the operator of the load during setting up. Two pull–out plungers were added to the front horizontal framing members (F361A). During normal operation the plungers are locked in their home positions by bayonet locks. During setting up they are pulled out by hand after the levers have been lifted and provide fixed resting support for the levers. Babbage would perhaps have simply jammed the levers in their raised positions.

There appears to be a dimensioning error on 177. A small section of shaft is shown projecting from the framing member to the lifting handle. The diameter of this shaft is drawn the same as the diameter of the bell crank boss. A bearing hole in the framing of that size would present a structural weakness. In addition, the outsized diameter of the boss for the lifting handle is shown smaller on 163 and on 161. It was assumed therefore that the shaft diameter on 177 shown projecting from the framing member is an error. The final dimensions are shown on E394.

*Modified Seventh Difference Lock*

The original design goes to some lengths to ensure that the figure wheels do not derange. Locks immobilise the figure wheels at appropriate stages in the cycle. During the carry portion of the cycle the figure wheels need to be free to receive carries and even here anti–deranging provision is made. Figure wheels are prevented from deranging during the carry cycle by horns on the carry levers which hold the figure wheels stationary in unwarned digit positions or advance the figure wheels one position in warned positions. So the figure wheels are secured at different times by locks and carry lever horns, or by engagement with the sector wheels during giving off.

However, the seventh difference axis has no even sector column alongside to the right. Consequently, during giving off even–to–odd, the locks are released and the first odd axis is unsecured. In addition, because the warning axis for the seventh difference column was omitted, the seventh difference column is unsecured during the carry cycle. The drive to the seventh difference lock was modified to secure the figure wheels during the parts of the cycle in which it is vulnerable to derangement.

The modification to the lock is based on the operation of a collapsible link shown in E22 and E31. The unlocking lever is pivoted on a bracket fixed to the right hand front upright. The unlocking lever, which is integral with the operating lever, is driven from the even sector bar. The even sectors are disengaged (even sector bar to the right) during giving off odd–to–even. In this position the slotted link holds the hinge member firm and the seventh difference lock is released, with the rest of the locks, by the odd lock bar lifting the lock acting through the twin–armed bell crank lever, the hinge member and the connecting rod. In the unmodified arrangement (i.e. with the bell crank driving the connecting rod directly) the odd lock bar would release the seventh difference lock and leave the column unsecured. During giving off even–to–odd and during the odd carry which follows, the even sectors are engaged and the even sector bar is to the left. With the additional mechanism, the slotted link releases the hinge member and when the odd lock bar moves left to lift the lock, the hinge member turns on the connecting rod pivot and fails to transmit the motion. This leaves the seventh difference figure wheel lock engaged as required, and held in place by its own weight.

**Vertical Motion Cams**

Drawings References: 159 (right), 160 (cam stack), 163, 168, 169.

Specifying the cam detail was the most taxing task in the overall specification of the engine and comparable in difficulty only to aspects of the printer design. Unlike the circular motions, which are intermittent but smooth, vertical motions are stepped i.e. lifting and lowering by fixed distances. The figure wheel axes, warning axes and locks are lifted and lowered by single fixed distances; the sector axes have a two step motion — with separate steps for partial and full engagement.

The following features of the vertical motion cams require specification: the height of the rises and falls to provide the required travel of the induced motions; the outside diameter of the cam blanks from which to manufacture the cams; the profile or shape of the rises and falls;

the dwell to sustain the motion for the appropriate duration; the timing of each motion – firstly to produce the correct start, duration and succession of the motions for a given cam pair, and secondly, to ensure that the motions from the cam pairs are correctly phased in relation to each other.

*Cam Follower Pivots*

The eight follower pivots are shown distributed around the circumference of a fixed pitch circle in 159. The exact size of the pitch circle is not explicitly given though it was assumed that Babbage would have chosen a convenient round dimension. Measuring the distance between the centre of the cam shaft and the centres of the cam follower pivots on 159 was not exact enough to recover the pitch circle radius with sufficient precision for manufacture and this radius was found through a combination of scaling and calculation. The X–Y co–ordinates of each of the pivot centres was measured referenced to the right–angled axes through the cam shaft centre. The radial distance from the cam shaft centre was then calculated as the hypotenuse of the triangle in each case and the list was inspected for a convenient round number. The X–Y coordinates of the even warning pivot, for example, were measured at 4.50" and 5 and 13/16th inches which gives 7.35086" for the radial distance. The figure of 7.35" was the number closest to a round number for the eight calculations and this was adopted as the pitch circle radius. Each of the pivot centres was then specified as the intersection of two loci – the pitch circle radius, and a coordinate referred to either of the right angled lines through the cam shaft centre.

*Modification to Pivot Positions*

A small change was made to the position of the even figure wheel cam follower pivot (159 lower right). As drawn, the cam follower boss fouls the horizontal bar for the even lock, and also fouls the end bar support (shown dotted). To avoid fouling, the position of this pivot was moved forward (down on the drawing) and slightly left to provide clearance. The slight alteration to the pivot entails lengthening the corresponding bar lever. As originally drawn there are only two different lengths for the eight bar levers. The two lower bar levers (even figure wheel and odd lock) are one length, and the remaining six bar levers, another. To maintain only two standard lengths of bar lever, the pivot for the odd lock cam follower was also moved to equalise the lengths of the two lower bar levers. The new positions were located on the same pitch circle. The cams themselves were modified slightly to compensate for the slightly extended lever lengths so as to ensure the correct final travel of the bars.

*Identifying the Cams*

For ease and consistency of reference the cams were numbered. The sequence starts with 1 at the bottom and runs through to 16 at the top (E21). The rotation of the cams viewed from above is anticlockwise. With the numbering indicated, the odd–numbered cams engage trailing levers and the even–numbered cams engage leading levers. The trailing levers are therefore below the leading levers on the follower pivots. The relationship between the numbered cams and vertical motions is as follows:

| | |
|---|---|
| 1 | odd figure wheel lift |
| 2 | odd figure wheel lower |
| 3 | even figure wheel lower |
| 4 | even figure wheel lift |
| 5 | even unlock |
| 6 | even lock |
| 7 | odd lock |
| 8 | odd unlock |
| 9 | odd sector lower |
| 10 | odd sector lift |
| 11 | even sector lower |
| 12 | even sector lift |
| 13 | odd warning lift |
| 14 | odd warning lower |
| 15 | even warning lift |
| 16 | even warning lower |

*Height of the Cam Rises*

The heights of the cam rises are specified by working back from the required vertical motions via the drive train to the cams. The travel of the vertical motions of the warning axes, figure wheel axes and sector axes are given on 171 where the length of travel is inscribed inside the circles representing the shafts: the vertical motion of the figure wheel axes is given as 0.3", that of the warning axes as 0.34", and that of the sector axes as 0.68" (for full engagement with odd and even figure wheels) and 0.34" for partial engagement when the figure wheels are restored. The significance of these figures is confirmed by the timing diagram (F385/1a) where these figures are reproduced alongside the arrows indicating the vertical motions.

The vertical travel required was multiplied by the ratio of the length of the arms of the bell crank levers to give the translational travel of the horizontal bars. The bar motion was then multiplied by the ratio of the length of the bar lever to the length of the follower arms to give the height of the rise (E327A, E371A for example). These calculated rises were then checked against the rises shown on 159 and 168. There were no significant corrections required.

Where the size of the required vertical motions is given explicitly as for the warning, figure wheel and sector wheel axes, this information was taken as the starting point of the calculation working back through the drive train to determine the cam rises. However, the vertical travel of the locks is not specified in the original drawings. The length of travel of the locks was derived from the depth of the figure wheel teeth and the gradient of the angled sliding bearing in the upper and lower bearing plates which support the locks. As before this figure was worked back through the drive train to determine the cam rise and then checked against the cam profiles shown on 169 (right) and the less clear view on 159 (bottom centre). Again there were no significant corrections required to the height of the lock rises though small adjustments were required to the dwell (see 'Timing' below).

*Cam Diameter*

The cams are shown with a standard outside diameter from which the outer shape deviates inwards by the height of the rises and falls (159, 169). The outer radius of the cams was taken as 6.46". This figure was calculated by triangulation and confirmed by scaling from 159. The pitch circle radius of the cam pivots is 7.35" and the length of the cam follower arms is 3.50" (see above). In the case of the sliding followers the follower arms are shown tangential to the outer cam circumference i.e at the top of the rise. The odd warning follower in 159 (top centre) provides one of the less cluttered examples. Here, as for the other sliding followers, the line joining the point of contact and the centre of the pivot is at right angles to the cam radius through the point of contact. In the case of the roller–cum–slider followers the same holds true i.e. the line through the point of contact of the roller and pivot centre is tangential to the outer cam circumference. The cam outer radius of 6.46" is thus calculated as the third side of the right–angled triangle formed by the radius of the pivot pitch circle (7.35"), the follower arm (3.50") and the cam radius. Scaling from 159 confirms that this figure was taken as standard for all the vertical motion cams. The figure of 6.46" was thus used to specify the maximum outer diameter of the vertical motion cam blanks (E391C).

**Design Alternatives**

Questions arise as to the implications of alternatives. The locus of the point of contact is an arc with radius equal to the effective length of the follower arm. In the case of non–conjugate cam pairs it is usual to specify the rises as bilateral deviations from a mean circumference rather than inward deviations from a standard outside dimension as Babbage has done. Using bilateral deviations from a mean circumference has the advantage of minimising the amount by which the arc needs to be taken into account in determining the actual angular position of the point of contact at the extremities of travel (i.e. the bottom and top of the rises). However, using deviations from a mean has the disadvantage of greatly complicating the specification. If mean circumferences were used in the case of conjugate pairs, the outside diameters of the cam pairs would differ. While there is no great penalty in this as regards manufacture, the loss of standardisation has a knock–on effect that introduces variations each of which would need to be catered for separately. For example, if the pivot centres are retained on a fixed diameter pitch circle, as Babbage shows, and the cam diameters varied, then the cam followers cease to be standard components: the lengths and angles of contact will vary from cam to cam and each cam pair would require additional calculation to specify its followers and ensure correct timing. Non–standard followers have the additional drawback of more expensive and troublesome manufacture. Another possibility would have been to abandon the fixed pitch circle for the pivot centres. In this case the radial distance of the pivot centres from the cam shaft centre would differ.
To retain tangential contact would again require non–standard follower arms. To disregard tangential contact would entail having to again take into account, in each individual case, the effects on timing of the arc traversed by the swing of the follower arms.

Babbage appears to have standardised the layout to simplify an already complex arrangement. Adopting a standard outside diameter for the cams, fixed pitch circle for the follower pivots, fixed follower arm lengths and tangential contact to the outer circumference, eliminated the need to cater for variant combinations of pivot position, contact angle, and arm length. The arrangement preferred by Babbage significantly reduces calculation and simplifies layout, manufacture and, moreover, reduces the risk of drafting and layout errors in attempting to keep account of variations. One example of the benefits of a standardised geometry arises in the determination of the cam keyways the positions of which are critical in ensuring the correct phasing of the various motions derived from the eight pairs of mating cams. With a fixed pivot pitch circle, standard follower arm lengths and standard outside cam diameters, the angular offset between contact points of leading and training arms can be taken as a standard 56°. This fixed offset between mating cams substantially simplifies the specification of the keyway positions for the full set of cams (see worked example below).

**Timing**

*Timing Diagram*

The basic timing data is derived from the original timing diagram F385/1a which provides the main source of information on the sequence and phasing of the motions in the calculating cycle. However, the original timing diagram lacks a level of fine detail and in some instances is in the nature of a guide rather than an exact specification. There appear to be a number of inconsistencies in notation (see separate section on 'Timing Diagram') though the most serious omissions concern the timing of the locks. Unlike the motions of the calculating axes and wheels, the action of the locks does not specified in a separate column of phased actions indexed against the cycle divisions. The locked and unlocked condition of the axes and wheels is indicated by an '**L**' (locked) and a reversed '**F**' (presumably 'free') though no detail is given for the lapping or phasing of the actions of the locks. For example, the circular motion of the odd figure wheel axis is shown commencing immediately the axis is lowered by 0.3" with '**L**' and '**F**' notations alongside the arrows indicating locked and unlocked conditions. The timing of the withdrawal or entry of the locks as an event with a finite duration is not indicated on the diagram and the avoidance of contention is simply implied.

In the case of the even figure wheels we find that the wheels are locked briefly between the end of the counter–clockwise motion giving off odd–to–even and the start of the carry cycle. Here, a unit interval of one Babbage–division (7.2°) is allowed for locking and unlocking. There is no horizontal grid on the diagram and the level of precision in the drafstmanship discourages exact scaling. The practice of allowing a single Babbage timing unit as the standard nominal interval for actions that are not specifically detailed is characteristic of the diagram. Non–specific timing gaps are not confined to locking action.

A new timing diagram was drawn (X21) which supplements the original diagram by providing the detailed lapping and phasing information otherwise lacking. As in the original diagram the redrawn version shows the timing of an event specified as the number of degrees of a 360° cycle starting from Babbage's original zero datum. Babbage's 50–division cycle is for convenience of reference but was not used in the modern specification.

*Timing: Lapping and Clearance*

In the case of leading lever rises the point in the cycle at which the motion commences coincides with the point of contact of the follower with the start of the rise. In these instances the start of the rise is taken as the critical reference. In other instances it is necessary to sustain a motion to overlap another motion. Here the start of the fall is taken as the starting reference and is used, making allowance for clearance, as the reference for the rise on the mating cam which initiates the return motion. For example, the odd figure wheel axis is raised and lowered once during each calculating cycle. Cam 1 lifts and Cam 2 lowers the axis. The lifting action on the cam occurs between 67° and 75° which gives the start of the leading rise and the duration. The corresponding falling rise on Cam 2 is given a few degrees clearance and is specified between 64° to 73° (E373A). The few degrees timing clearance correspond to a clearance of about 0.003" between the passive follower and the cam [check]. The action to lower the axis is determined by the rise on Cam 2. This action occurs between 357° and 5° and a corresponding clearance is allowed on the falling rise on Cam 1 (E374A). The timings given in this example are taken from the cam specification. These are not always identical to the timings on the redrawn timing diagram. For example, the timing diagram (X21) shows the lifting action commencing at 68° though the start of the rise on the cam is shown at 67°. The foot of the follower is slightly angled and as the foot starts to mount the rise the actual point of contact leads the point of contact otherwise operative on the plateaus. A 1° lag in the cam rise is introduced to compensate.

Allowance was made for the roller followers used for the sectors. The active surface leads the centre of the roller by a few degrees and this was taken into account when specifying the take–off point of the rise.

*Timing: Locks*

The cams for the locks presented special difficulties. A single calculating cycle requires four separate episodes of engagement of the locks. Three short engagements are specified for correction of minor derangements and to secure the figure wheels during momentary otherwise unsecured intervals. The longer locking period is to secure the figure wheels after giving off while the sectors disengage and the alternate axes carry. The figure wheels that have just given off remain locked until the sectors commence restoration.

Where allowance is made for the entry and withdrawal of the locks on the original timing diagram (F385/1a), a nominal single Babbage timing unit is allocated i.e. an interval of 7.2° is allowed for locking and unlocking. However, working back from the rises on the locking cams shown on 169 the actual period locking and unlocking would occupy is 10° (4° for entry, 2° for dwell, and 4° for withdrawal). It is clear from the cam details on 169 and 159 which specify the height of the rise, pressure angle, dwell and fall that Babbage did consider in detail how the operation of the locks should be phased. However, the detailed mapping from timing to cam specification is omitted in the original drawings. In the modern implementation a minimum allowed was 11° to ease the pressure angle (4° plus 3° plus 4°). The difficulty was then to fit these slightly stretched locking actions (particularly the three short engagements) into the already crowded and therefore tight timing cycle. The solution in the case of the figure wheel axes was to shorten the duration of the circular motions (without reducing the total rotation) to allow for the locking action – in particular for the withdrawal of the locks so as to avoid contention between the lock and the circular motion that follows. The dwell of the locks was extended slightly in some instances to cater for the arc of the point of contact of the follower.

**Phasing the Vertical Motions**

The cams are keyed to the central cam drive shaft (Cam 1 is the sole exception, see previous section). The position of the keys is the critical determinant in ensuring the correct phasing of the motions separately created by the eight cam pairs. The original drawings do not indicate a datum for the keyways nor any keyway positions. An arbitrary datum was chosen: the single keyway running the length of the cam drive shaft is positioned at the front of the engine when the cycle is at 0°. For drafting convenience cams are drawn with the

0°–position of leading lever cams at 12–o'clock.

If the active point of contact on each of the follower arms was at the front of the engine then all the keyways would be in line and at zero. However, the pivots are distributed around the cams and the cam keyways need to be offset relative to the front of the engine to compensate i.e. each cam needs to be rotated from the zero datum until the event on the cam that is to occur at zero is at the point of contact of the appropriate follower. The position of the cam keyway is then fixed at the front of the engine. Because of the standardised geometry of the pivots, follower arms and standard outside cam diameter, once the offset calculation is done for one cam of the pair, the keyway of the mating cam is found by adding or subtracting a fixed offset of 56° depending on whether the mating cam is leading or lagging.

The offset for each of the 16 cams needed to be calculated separately. The keyway calculation for the even warning cams (Cams 15 and 16) is shown in the example (see Fig. 1).

*Keyway Offset — Example*

*Figure 1a.* Keyway Offset Calculation for Odd and Even Warnings
*Figure 1b.* Locus of Point of Contact

The first step is to calculate the angular displacement (Θ) of the pivot centre. The coordinates of the even warning pivot (Fig. 1a) are given by the pitch circle radius (7.35") and the distance from the horizontal axis (4.50" scaled from 159) (see 'Cam Follower Pivots: Position', above). The displacement of the pivot centre is simply given by *arcsine* 4.5/7.35.

The next step is to determine the angular displacement of the point of contact of the follower arm. Since the cams rotate anticlockwise, the leading arm is the one to the right of the pivot and the trailing arm is to the left. With the leading arm tangential to the outside diameter of the cam (i.e. at the top of the 0.50" rise) the angle subtended at the cam shaft centre by a 3.50" follower arm is 28°26'. It would be convenient if this could be rounded down to 28° to save carrying the 26' through each calculation. For practical purposes the error introduced by this rounding down was ignored on the following considerations. The height of the rises in the case of the warning cams is 0.5" (see Fig. 1b). The foot of the active rise determines the start of the motion for both leading and trailing arms. The locus of the point of contact is a circular arc of radius equal to the effective length of the follower arm. Taking this into account Fig. 1b shows that a 3.50" arm at the foot of the 0.50" rise subtends and angle of 28°5' i.e. only 5' of arc off the rounded figure. Working backwards it emerges that a 3.45" arm at the top of the rise subtends and angle of exactly 28°. If for purposes of calculation the distance to the point of contact is taken as 3.45" (the physical distance remains unchanged at 3.50") then the difference on the outer circumference of the cam amounts to 0.050" which corresponds to a worst case timing error of 26' of arc in the start of the falls and the end of the rises.

Using a 3.45" arm in the calculation to represent a physical arm of 3.50" (it is emphasised that the physical arm remains 3.50") introduces timing error of 5' of arc for a 3.50" arm at the foot of the rise (a 3.45" arm tangential at the top or a rise has the same timing relationship as a 3.50" arm at the foot of a 0.59" rise). A worst case error of 0.050" on the circumference of a 13" cam was considered to be acceptable and a worst case timing error
of 26' was considered to be an acceptable price to pay for substantial simplification of both calculation and manufacturing specification given that the resolution of the timing had already been increased from Babbage's 50–division per cycle scale to a more conventional 360° scale (a factor of 7.2). The errors introduced by using 3.45" as the arm length for purposes of calculation, and taking both follower arms tangential to the outer circumference at the same time (physically this never occurs as one follower is at the foot of a rise when the other is at the top), were therefore regarded as negligible for practical purposes in all cases except the lock timing for which the timing cycle is particularly tight and for which special provision was made for the circular locus of the point of contact.

With the follower arms represented by a 3.45" tangential to the outer circumference we are now in a position to determine the angular displacement of the points of contact i.e. $\Theta \pm 28°$. The cams rotate anticlockwise viewed from above. The even warning pivot thus lags the zero position by $270°–\Theta$ and the leading point of contact lags by $270°–\Theta–28°$. The later the event in the timing cycle the further clockwise on the cam the corresponding rise or fall i.e. lag corresponds to clockwise displacement of the keyway. The keyway offset for the leading even warning cam (Cam 16) is therefore $270°–\Theta–28°$ clockwise from the zero datum (204°15'). Similarly, the displacement of the trailing point of contact lags the leading point of contact by round and convenient 56°. This translates into a clockwise displacement of keyway for the trailing cam (Cam 15) of 260°15'.

*Separation of the Cams*

All the vertical motion cams were made from identical blanks (E391C) specified with maximum metal and machined to suit differing requirements of outer shape and vertical spacing. Fourteen of the cams (1 through 9, 13 through 16, plus 11) have the bosses machined down for the six closely spaced pairs. The two cam pairs for the sectors which are spaced further apart have the bosses left intact (10 and 12). Cams 1 and 2 (odd figure wheel axes) have the standard close spacing but special provision is made to accommodate the lubricated annular cam shaft bearing set into the lower framing plate. 160 shows the bearing and the cam in the same plane but no details are given for clearance or for securing the lower–most cam (Cam 1). The boss on Cam 1 was removed to clear and the cam shaft bearing and Cam 1 is screwed and dowelled to Cam 2. The lower cam shaft bearing prevents Cam 1 being keyed to the shaft. Cam 1, fixed as an undercarriage to Cam 2, is driven by Cam 2 which is keyed to the cam shaft. Each of the other fourteen cams is each keyed to the cam shaft and each cam is screwed and dowelled to its mating cam. Cam 1 is the only cam not keyed to the shaft.

*Verification*

The cam rises and phasing were verified graphically. Tracings of the cams were overlaid on the mating cam and as though fixed together and the motion and clearances checked using tracings of the follower arms. The standard 56° lag is a significant drafting convenience in ensuring that the dowelling and fixing holes of the mating cams line up as well as aligning the aperture cutouts. (See E383A&B for warning cams (leading); E384A&B for warning cams (trailing).

The cams were manufactured without keyways in the first instance. During assembly each cam was fixed and pinned to its mating cam and turned by hand on the shaft. When the motions of the followers were verified, the keyways were cut with the cams still fixed in mating pairs.

*Manufacture and Assembly of the Follower Arms*

The original drawings give no details of how the pairs of follower arms are to be fashioned. The two roller–cum–slider arms were made as one piece (E353&4). The twelve twin–slider arms were made in two parts spigotted together so that the two arms could be rotated to alter the angle between them. The paired arms were assembled on the pivot shaft with the position of the arms provisionally fixed with grub screws. Fine adjustment to the arm positions was carried out with the assembly and related cams in situ. The follower assembly was then removed from the cam stack, the arms (still fixed in relation to each other by grub screws) were slid off the pivot shaft, welded together, slid back on and then pinned to the pivot shaft. There was an awareness that this procedure might be judged to be overcautious. It was nonetheless considered preferable to having to scrap out–of–specification follower arm assemblies that might result from fixing the angles without trial.

The contact foot of each slider arm was case hardened prior to adjustment and welding to ensure that any heat distortion from the hardening process did not affect the trial settings.

## Appendix

[**Note**: Some of this material has been incorporated into separate sections in the Vertical Motion material and is repeated below].

*Timing Diagram* BAB [F] 385/1a

This drawing is the main source of information describing the sequence and phasing of the various motions in the calculating cycle. In general single arrows are used to denote circular motion (up arrow clockwise and down arrow counter–clockwise); double headed arrows are in general used to denote vertical motions with the length of travel of the motion indicated alongside in inches. However the convention of double and single arrows for vertical and circular motions is not consistent. For example, the lowering of the odd sector axis by 0.68" is indicated by a downward single–headed arrow, the succeeding upward motion of 0.34" is denoted by an upward single–headed arrow, and the final restoration of 0.34" is indicated by a double–headed arrow. There are also evident errors within the convention. For example, the direction of rotation if the odd sectors, figure wheels and carries is incorrect, as well as those indicated for the even warning and carry levers.

Broken lines indicate that the motion is not necessarily sustained for the full duration of the cycle interval allowed. For example, the circular motions of the figure wheels are indicated by broken lines. Here the duration of the motion is data dependent i.e. the proportion of the figure wheel axis rotation that the that the figure wheel is driven depends on the number–value of each figure wheel. Not all the notations were decoded. For example, the significance of the presence or omission of a bar on the tail of an arrow is not clear.

The timing diagram lacks a level of fine detail and in some instances is no more than a guide. The most serious omissions are in respect of the timing of the locking and unlocking actions. Unlike the motions of the calculating axes and wheels, the action of the locks is not represented as a separate set of phased actions indexed against the cycle divisions. The operation of locks, for example, is indicated by a bold '**L**' to indicate engagement (locked) and by a reversed '**F** (presumably to indicate 'free') but no close detail is given for the lapping or phasing of the entry and withdrawal of locks to avoid contention between conflicting motions. Taking the even difference figure wheels as an example we find that at the wheels are locked briefly between the end of the counter–clockwise motion from giving off odd–to–even and the start of the carry cycle. A unit gap of one Babbage–division (50 to the cycle i.e. 1 division corresponds to 7.2°) is allowed for locking and unlocking. Though there is no horizontal grid on the diagram and the precision of the draughtsmanship discourages exact scaling these one–division gaps appear to be the nominal standard interval allowed for motions that are not contiguous in time.

However, working back from the locking cams (169) indicates that the actual period that locking and unlocking would occupy is 10° (4° for entry, 2° for dwell and 4° for withdrawal). In the modern implementation the minimum allowed was 11° (4° plus 3° plus 4°) but the dwell was varied slightly in some instances.

The '**L**' and '**F**' notations thus indicate locking status at the start of the specified motion ('locked' or 'free'), and the completion of the locking or unlocking action in time to avoid conflict is implied. It is clear from the cam details on 169 and 159 which specify the height of the rise, pressure angle, dwell and fall that Babbage did consider in detail how the operation of the locks should be phased. As far as fine detail is concerned the timing diagram is evidently representational with the detailed mapping from timing to cam specification omitted. All the cams were specified and verified from scratch. There were no significant deviations from Babbage's specifications.

A new more detailed timing diagram was drawn [Drawing Reference X21] to specify more exactly the cycle timing and to include detailed phasing and lapping of the various motions. Babbage's arrow notation was retained and notational inconsistencies in the original were eliminated. New notations were added: two hatched lines on the figure wheel circular motion arrows indicate addition – this to distinguish between figure wheel addition and return motions; horizontal arrows that traverse the timing diagram columns indicate causal trains; and diagonal arrows indicate secondary warnings produced by the carry cycle.

## Drive

Drawing References:
Main drawings are 159 and 163. Additional views: 160 shows the cam drive; 164 shows the hand crank; 165 shows the intermittent drive for the carry axis; 167 shows the printer drive.

The 'drive' refers to the system for transferring power from the hand crank to the cams, printer and carry axes.

### Description

Power is transferred from the hand crank to the cam stack drive shaft via bevel gears at the top of the cam shaft (163). A ratchet is located to the right of the drive shaft bearing and ensures that the drive cannot be reversed. The ratchet drive is coupled to the bevel gear via a rocker clutch operated by a gut cable running from the printer. The purpose of the clutch is to disengage the drive immediately a page of printed results is completed, without any risk of overrun. The intention is to halt the calculation at an exact point in the calculating cycle to allow the stock receiving the printed impressions to be renewed. Activating the clutch uncouples the drive. The engine halts and arrests the figure wheels is a state ready for the commencement of the next run of results.

Each of the seven carry axes is driven by a bevel gears pinned to a horizontal shaft driven by an intermittent drive mechanism (165 lower right). The bevel drive for the first odd axis is omitted as the seventh difference carry mechanism was regarded as redundant. The shaft coupling the bevel gear and the intermittent drive mechanism is hollow. The shaft from the handle crank passes through this hollow drive socket and is supported by the bearing on the right hand lower end frame.

The printer drive shaft which runs the length of the underside of the engine is driven by a bevel gear at the lower end of the cam shaft.

*Crank*

The crank consists of a drive arm and a drive handle. The handle is made from lignum vitae which, though not specified, was regarded as typical. The handle is free to turn on a shaft which is peened over at each end. A thrust washer is fitted at the outer end.
No end elevation of the crank is given so the intended shape of the arm is not known.
The chosen shape was taken as characteristic. The length of the arm is taken from 163 and 164, and the thickness from 159, 163 and 164.

*Ratchet Drive*

The ratchet prevents the engine being driven in reverse which would damage the calculating mechanism. The ratchet allows the operator to start turning from a stationary position best suited for the exertion of controlled effort. The ratchet also allows the operator to nudge the engine to a particular point in the calculation cycle by a series of short pulling movements.

The ratchet mechanism is shown on 159 lower right. It consists of a ratchet wheel and matching pawl housed by a pawl wheel which acts as its casing. A leaf spring fixed to the pawl acts against the inside of the case to provide the return action. The pivoted end of the pawl nestles against a machined pocket in the side wall of the case. The machined pocket takes the drive load rather than the pawl pivot which is a loose fit. The pawl is sandwiched in place by the pawl wheel cover which is rebated into the case and secured with counterbored cover screws. (159 shows the case with the cover removed but the fixing screws fitted.)

**Implementation**

The counterbores for the pawl wheel cover fixing screws have insufficient clearance from the wall of the casing. The section view (159) shows the cover fixing screws breaking through which is uncharacteristic. Fixing screws were therefore reduced from 5/16" to 1/4" BSW, the head reduced to 7/16" and the length was shortened to avoid breakthrough. The case is cast and machined.

The outer wall of the case was extended to take 56 gear teeth to allow a four–to–one reduction in the drive. The addition of reduction gearing was in response to concern that an operator would be unable to exert sufficient force to turn the engine over. The additional parts for reduction gearing are a 16–teeth pinion (D323), a crank (D322) and a horseshoe mounting (D321) for the pinion. The crank, D322, is a repeat of the pawl wheel crank. The crank is keyed to the pinion, and the drive arm is fixed to the crank with a shoulder screw (D327). The mounting is fixed to the underside of the cam stack upper mounting plate. Using the reduction gear is optional: the handle crank can be removed by undoing the shoulder screw and refixing the drive arm directly to the ratchet wheel crank, as Babbage intended.

Without the reduction gears the engine is driven by turning the handle anticlockwise as seen facing the status plate (otherwise called the chapter disc) and the addition of the reduction gearing reverses the direction of the drive. The main loads on the drive occur at the start of the cycle and at the half–cycle. These are caused by all the figure wheel and sector wheel locks coming out simultaneously. There are thus two short peak loads in one full cycle. Lesser cyclic load occur at the start of the carry portion of the cycle i.e. at 10 degrees CB (72 degrees) and 35 degrees CB (252 degrees).

For the convenience of the operator the handle is positioned at the lowest point of its travel for a pulling action at roughly waist height to overcome the start–of–cycle load. Without the reduction gears the half–cycle load would then occur at the top of the arc requiring a pushing action which is awkward and could reasonably have induced Babbage to include a reduction gear of his own. An additional advantage of the reduction gearing is that the full– and half–cycle loads both occur with the handle at the bottom of its travel and can be overcome by a pulling action which is more convenient than with a single–turn cycle.

*Clutch*

The clutch is shown in end elevation (159 lower left) and in section (159 centre left). The clutch rocker has a short lug and an extended lug protruding from the ends of the basic oval shape of the rocker and which rock in slots machined into the rim which stands proud of the clutch body. The extended lug protrudes beyond the clutch body and, with the clutch engaged, inserts between a pair of lugs on the clutch bevel. In 159 (centre left) the clutch is engaged with the clutch rocker shown vertical and disengaged in the inclined position.

To disengage the drive, the scoop lever, operated from cable from the printer, positions the scoop cam (shown as dotted position in 159 centre left) so as to drive the extended lug clear of the driven lug when next it traverses the bottom of its trajectory. The extended lug engages with the inclined face of the scoop cam, drives the scoop lever fully home (position shown slotted on 159 centre left), and is itself driven out of engagement with the driven lug so breaking the drive.

*Clutch Operation – Difficulties*

The clutch rocker is likely to overrun the scoop cam after disengagement following the sudden release of load. Re–engaging the clutch is problematic with the arrangement as drawn. To re–engage the drive the clutch rocker is rotated to bring the extended lug opposite the bevel lugs with a view to reinsertion. However, with the rocker out of engagement (inclined position) the extended lug clears the first of the two bevel lugs but is prevented from re–engaging by the scoop cam. If the scoop lever is reset (moved to the position with the scoop cam shown solid) re–engagement is similarly obstructed. Setting the rocker to the engaged position (vertical in 159 centre left) while outside the two bevel lugs causes other difficulties. As the rocker approaches the scoop cam position the extended lug will fail to clear the bevel lug and will engage with the *outside* face of the trailing lug (the trailing lug is the upper of the two bevel lugs as shown on 159 lower left and is the first one met by the extended lug).

If Babbage had intended the bevel to be driven from the outer face of the lug then it is likely that he would have drawn a radial face to ensure a flat bearing surface. Drawn as it is with a square shoulder, the bearing surface is a sharp edge which is uncharacteristic. A further difficulty driving the engine from the outer face of the bevel lug is that the timing of subsequent disengagements would be delayed i.e. the engine would halt in a slightly more advanced part of the calculating cycle because the extended lug would be lagging behind its intended position between the two bevel lugs. The sequence for re–engagement would be to rotate the rocker to clear the scoop cam, reset the scoop lever, reset the rocker into the engaged position while clear of the bevel lugs, rotate the rocker till the extended lug engages with the outer face of the trailing bevel lug, drive the bevel clear of the scoop cam area, halt the drive, disengage the rocker and re–engage the extended lug between the two bevel lugs, continue with the calculation run. This an unlikely sequence and contra–indicated by the square shoulder of the leading bevel lug. The omission of provision for re–engaging the clutch seems to be an oversight.

Since the ratchet ensures that the drive is unidirectional the trailing bevel lug is redundant. The only situation in which the trailing lug would come into play would be to reverse the drive and there is no means in evidence for disabling the ratchet to allow this. Since there is no functional reason for the trailing bevel lug this was omitted. With only the leading lug present, re–engaging the clutch is simply achieved by returning the rocker to the engaged position once clear of the scoop cam area, resetting the scoop lever and rotating the drive until the extended lug meets the driven lug on the bevel. The single remaining bevel lug was strengthened by making the outer face radial rather than parallel to avoid a weak corner when casting.

During commissioning and debugging it was found that a useful way of unlocking jams was to reverse the drive, and a means of doing this using the hand crank would have been useful. In the event, reversing was accomplished by gripping the cams and backing the engine off by turning the cam stack a short distance in reverse. It is difficult to back the engine off more than a very short amount by gripping and turning the cams because of the load and the awkwardness of the movement. Backing off by hand in this way was considered preferable to disabling the ratchet drive as prolonged reverse drive of inadvertent reverse drive at an inappropriate point in the calculating cycle could damage the calculating mechanism.

The views of the extended lug are inconclusive with respect to the shape of the area that engages with and slides up the scoop cam. The end of the extended lug was radiused to present a larger contact surface with the scoop cam.

The scoop lever is pulled in by a gut cable from the printer and is reset by hand to resume calculation. With the clutch engaged there is no provision for positively locating the scoop lever which rests free against the slack of the cable. The risk of the scoop lever drifting into unwanted engagement or into an end–on collision with the extended lug of the clutch rocker was considered to be real. A counterbalance was added to provide positive resistance to the printer cable. The counterbalance maintains cable tension which prevents the printer cable jumping its pulleys. It also resists the tendency for the scoop lever to be pulled in unintentionally by the weight of the printer cable lever, by any elastic tension in the gut, or by any tendency of the gut to curl. The counterbalance is provided by the addition of a weight suspended over a pulley by a gut cable attached to an eyelet added to the opposite end of the scoop lever. The pulley is fixed to the rear horizontal framing member and is a repeat of the printer cable pulley.

*Disengaging the Clutch*

The clutch is disengaged by the printer when a page of stereotyping has been completed. 163 (bottom left) shows a cylindrical weight resting on a half–round gutter. The trip lever is linked to the scoop cam by a gut cord which runs over the two pulleys. At the end of a stereotyped page the release of the weight operates the trip lever which pulls the gut cord and sets the scoop lever to disengage the clutch on the next pass of the extended lug (dotted position on 159 centre).

Strictly speaking, 163 is indeterminate with respect to the shape of the weight: it is unclear whether a sphere or cylinder is intended. The representation of the receptacle as a gutter rather than a ring suggests a cylinder. This is confirmed by 166 which shows a freely suspended rod. The view shown on 166 omits the base supports. Drawing these in shows that the right hand end of the cylinder as drawn will foul the base support. The left hand end of the weight, as drawn, would foul the rewind clutch during operation. If a cylindrical weight were to be used it would have to be shortened to avoid fouling.

There is a further concern about using a cylindrical weight. There are two basic formats for stereotyped results: generating successive tabular results below one another and starting the second column after completing the first (line–to–line movement); and generating successive entries alongside each other, two to a line (column–to–column movement). In both instances the weight is raised above the clear of the sides of the gutter in its unreleased position. Without the sides of the gutter to constrain the weight it seemed likely that the cylinder would tend to twist and fail to align with the gutter on release.

In view of these two concerns the cylindrical weight was replaced by a spherical bob and the gutter by a ring–shaped hoop. The spherical arrangement allows the weight to rise clear of the ring and relocate on release regardless of any twisting. The use of a spherical bob also solves the problem of the cylindrical weight fouling the base support and rewind clutch.

The lever is shown in elevation only (163) and no further original details are supplied. The lever was made in two parts – the completion trip which includes the ring and ring arm, and the trip lever which has the eyelet for the gut cord. Both are fixed to the same shaft but the angle between the trip lever and the completion trip can be adjusted by means of a grub screw so as to take up any slack in the gut cord while maintaining the height of the completion trip.

*Status Indication*

No provision is made in the original drawings for keeping track of or indicating the position in the calculating cycle at any time. Marking the upper–most cam was considered. This was rejected as viewing is partially obstructed by the cam stack framing plates. Instead a status plate or chapter disc (so called because [ask Turvey]) was fixed to the smooth (non–ribbed) side of the phasing wheel which is in direct sight of the operator. A single rotation of the phasing wheel corresponds to one full calculating cycle of the machine. The brass chapter disc is engraved every five divisions and follows Babbage's fifty–divisions–to–the–circle convention. Engraved labels indicate main positions in the cycle: "Full Cycle" (0); "Half Cycle" (25); "Set Odd" (20); and "Set Even" (45). A swan–necked pointer is fixed to the right front upright and acts as a fixed reference. Status indication is essential for the setting up procedure. For example, at 10 and 35 divisions the odd and even axes respectively are zeroised and the setting locks inserted. The chapter disc and pointer provides this necessary position information.

**Carry Shaft Drive**

Drawing References:
Main reference 165 bottom right. Also 159 top left; 160 top, centre right shows sprung roller; and 163.

*Description*

The ripple–through or successive carry is performed by a helical arrangement of carry arms keyed to the carry shafts. The carry shafts are driven by bevel gears pinned to the horizontal drive shaft. 163 shows eight carry shaft bevel drives. The first odd carry mechanism (extreme right) was considered redundant, and omitted. The intermittent rotary drive for the carry drive shaft is provided by the mechanism shown on 165 lower right. This consists of a large phasing gear with incomplete sets of teeth and a twin tooth drive. The twin tooth drive is fixed proud of the plane of the gear and meshes with the single impact tooth fixed to the register pinion. The register pinion is fixed to a register wheel which has a circular detent for a sprung roller. The spring tube, register arm and roller are shown on to the top of 160, centre right. The detent and roller holds the pinion during the idle periods in the correct position for remeshing with the phasing gear at the start of the next carry. There are two carry episodes in one full calculating cycle – one for the odd axes and one for the even. The continuous rotary motion of the phasing gear, driven by the hand crank via the ratchet drive and clutch, is converted into intermittent rotary motion of the register pinion to provide two episodes of carry shaft rotation for each full calculating cycle. The odd and even carry shafts all rotate together though only one set is active during a given episode.

The arrangement on 165 lower right shows the phasing gear and pinion positioned at the start of a carry. The two impact teeth mesh and in turn ensure correct meshing of the drive teeth. The first set of twenty teeth on the phasing gear, drive the register pinion full circle. The last tooth on the pinion is incomplete: it is angularly truncated to clear the trailing tooth of the phasing gear so as to leave the register pinion in its idle position. The register pinion idles during the toothless portion of the phasing wheel's rotation and the second carry commences when impact teeth next mesh. Each carry occupies 104 degrees of a 360 degree cycle, and is followed by a 76 degree idle between carries.

The function of the impact teeth is to provide correct meshing of the drive teeth when engaging and to allow the use of materials tough and durable enough to withstand repeated impact. The phasing gear and register wheel are cast; the impact tooth and twin tooth drive are machined from high–grade (EN16M) steel. The twin tooth drive is fixed only at two positions near the rim of the phasing gear and its cylindrical centre stands clear of the phasing wheel boss . The effect is to provide a form of shock absorbtion through the sprung action along the length of the twin tooth drive.

The impact tooth and twin tooth drive are shown fixed with screws though no further details given. To spare the threads of the fixing screws taking the impact, the fixing holes were fitted with impact sleeves which have the action of doweling the impact tooth and the twin tooth drive in the register pinion and to the phasing gear respectively. The sleeves are fixed with cheese head screws. (The impact sleeves, D398, were added during the build, Red Book 22.11.90)

*Phasing Gear Key Position*

It was noticed that the line through the centre of the phasing gear and the mid–point of the keyway deviates by 1 degree from the line bisecting the angle between the lower pair of webs (165 lower right). This is a subtle and curious deviation and its possible significance takes a little interpreting. The phasing gear is driven by the bevel drive and drive socket which is hollow to allow for the drive shaft. Pinning the bevel and phasing gear would be ill–advised and the original drawings show the phasing gear keyed (165). The position of the key for the clutch rocker is shown (159) opposite the extended lug, but no position is shown for the keying between the bevel gear and drive socket. The position for the bevel gear keyway was assumed to be in line with the keyway for the clutch rocker. This assumption was made on the basis that if the position of the bevel gear keyway was other than arbitrary, it would have been positively specified. It was assumed that the keyways in the drive socket for the phasing gear and the bevel gear would be in line given the additional trouble of machining with an offset. So the argument is that the bevel and phasing gears need to be keyed because they cannot be pinned, that the bevel gear keyway is in line with the clutch rocker keyway, and that the drive socket keyways should be in line for convenience of manufacture. Working back from the clutch and drive bevel in this way fixes the position of the key for the phasing gear.

Working from the register pinion end it is evident that the relative positions of the shafts for the phasing gear and the register pinion are fixed by the layout of the calculating mechanism and the drive. In 165, which shows the configuration at the start of a carry, the position of the twin tooth drive is determined by the geometry of the impact teeth and the requirements for correct phasing of the intermittent carry drive. The twin tooth drive needs a web to fix to for support. This then fixes the orientation of the webs and therefore of the phasing gear. So the orientation of the phasing gear is determined from the register pinion end, and the position of the phasing gear key is determined independently by working back from the clutch and bevel. There is therefore no reason why the centre line of the keyway should coincide with the line bisecting the lower two webs. The fact that the discrepancy is as little as 1° is an additional confusion: there is no reason in principle why it could not have been something more substantial and therefore more noticeable.

In many of the drawings gear teeth are characteristically referenced from the flank of the tooth rather than from the centre or from the inter–tooth gap. The occurrence of the 1° difference is not helped by the fact that the centre line through the keyway closely coincides with the line taken from the flank of the opposite tooth.

If the keyway in the phasing gear were machined to preserve symmetry i.e. so that the centre–line through the keyway bisected the angle of the two lower webs, this would amount to ignoring the 1° offset. The implications of this would be to retard the carry timing which brings the end of the carry part of the cycle critically close to the point at which the locks come in. The 1° displacement of the keyway was strictly adhered to on the assumption that the offset was deliberate and dictated by timing needs.

*Lubrication of Cam Shaft Bearings*

The cam shaft is held at the top by the upper framing plate journal bearing and is supported by a thrust bearing in the lower framing plate (160 and 163). Detail of the thrust bearing is shown in 160 lower right. The bearing consists of an inverted cup–shaped collar supporting a shoulder on the cam shaft. The collar, which is keyed to the shaft, rests in a annular oil bath located in the lower framing plate. No provision is made for lubricating the cam shaft bearings: access to the annular oil bath is obstructed by the closely spaced cams in the lower section of the cam stack, and the upper bevel gear obstructs access to the upper journal bearing.

Oil is supplied to the bearings through an oiling hole in the upper bevel gear and conveyed along the length of the cam shaft to the thrust bearing through a channel machined into the shaft opposite the driving keyway. Oil collects in an unsighted oil cup fitted to the upper framing plate below the bevel gear. This acts as a reservoir and prevents oil spreading over the upper framing plate. The oiling channel in the shaft starts off shallow and deepens to full depth below the upper framing plate to assist flow. The shallow channel rotates in the oil cup and drains oil into the channel lubricating the upper journal bearing at the same time. Oil drains down the channel and collects in the lower oil bath which forms the lower half of the thrust bearing. Internal overflow lubricates the lower journal bearing. To ensure that oil is properly distributed over the bearing surface between the thrust collar and the annular bath, two D–shaped cutouts were machined into the thrust face of the collar.

## Framing

Drawing References:

| | |
|---|---|
| BAB [A] 161 | General Plan and Detail of the Calculating Part (See Elevation "L") |
| BAB [A] 163 | Elevation of Difference Engine No. 2, 1/4 size |
| BAB [A] 164 | Plan of Difference Engine No. 2 (1/4 size) |
| BAB [A] 176 | Plan of Calculating Part of Difference Engine with the means of conveying numbers to Stereotype sectors |

*General*

'Framing ' refers to the fixed structure supporting the working parts of the engine. Framing includes the two base supports which run the full length of the engine and on which the engine rests, vertical framing supports, upper and lower horizontal frames, bearing plates supporting the calculating axes, printer shaft mountings which straddle the two base supports, rack mountings, and the cam stack pillars, fixed cam stack framing plates, cam shaft thrust bearing, and the drive shaft mounting.

*Description*

The calculating axes are contained between bearing plates which span, front to rear, the upper and lower horizontal frames. The upper frame is shown resting on the uprights (163) and the lower frame (roughly midway up the engine) is shown with open corners let in to the uprights (161, 176). The upper frame is shown as a single tray (164), presumably a one piece casting. There is no indication on the elevation (163) of piece part assembly of the frame. Similarly, the lower tray is shown as a one–piece part with no indication of sub–assembly (161, 176). The bearing plates rest on machined surfaces running the full length of the frames (176 shows lower frame; also tracings B6. No fixing is shown for the upper (163) or lower frames (161, 176) to the uprights. (The fixing on the top left of 161 is for the rack support.) Positions for the fixing holes for the bearing plates are also not shown.

There is an inconsistency in the dimensions given for the overall length of the frame as annotated at the bottom of 161 and the dimension scaled from the arrangement as drawn and calculated from the geometry of the calculating mechanism. The distance between figure wheel axes derived from 171 is 6.98 inches [see section C]. Seven such pitches give the distance between the first and last figure wheel axes as 48.86 inches Adding to this the scaled dimensions to each end of the frame (5 inches to the right and 8.14 inches to the left) gives a figure of 62 inches for the overall length of the frame. The annotation at the bottom of 161 gives 62.25 inches for this same overall dimension – a discrepancy of 0.25 inches. The scaled figure of 62 inches was used and the annotation ignored. (A similar inconsistency between annotation and scaled dimension occurs in the specification of the drive shaft bearing shown on 159 and 163. In this case the annotation was regarded as superseding the drafted version.).

**Implementation**

Drawing Reference: 377F31

| | |
|---|---|
| Frame Ends (Left and Right): | 311 A–D |
| Framing Pieces (Front and Rear): | 312 A–D |

Because of the difficulty casting large one–piece framing pieces, and the difficulty machining the surfaces for the bearing plates, especially on the upper frame, the two framing trays were each made from four separate parts: a pair of end pieces (short dimension of the engine, left and right), and a pair of members (long dimension, front and rear).

The original drawings do not show an end elevation. Curves were added to the upper and lower end pieces modelled on those shown for the front lower member (163). Apart from appearances, the additional material increases the shortest diagonal dimension and strengthens the corner.

The four end pieces are made from one pattern. All four pieces are different. The three other frame ends are cast from the same pattern without loose pieces and differently machined. The upper and lower frame ends have different vertical depths; the lower right hand frame end (B) takes the main drive bearing and loose pieces are added to the pattern to provide for this; the top left frame end (D) has mounting holes for the printing mechanism, a central circular clearance cutout, and a rectangular cutout. The need for clearance cutouts for the printer (D only) became evident after the drawings had been sent out for manufacture and while the printer drawings were being progressed. The frame ends were sent for additional machining after they were first delivered.

No provision is made in the original design for securing the engine while being transported. Additional fixings were provided on the top frame ends to take transportation bars. Eight tie–bars strap the framing to the steel base on which the engine rests. The transportation fixings are omitted on the lower frame ends by removing the loose pieces from the pattern when casting. The transportation bosses are not visible to normal viewing from ground level and can only be seen if the machine is viewed from above.

*Front and Rear Framing Pieces*

The bearing plates span the front and rear framing members and are fixed to a raised machined surface running along the top of the inside bottom flange of the angle (tracings B6). The height of this machined surface was taken as 1/8th inch. This dimension is taken from the depth of the internal pad on the upright intended to receive the lower frame (176). Pads for pulley brackets are required on right hand side only, front and rear. Only these two pads were machined. Redundant pads on left front and rear are unsighted and were left intact after casting.

The four long framing pieces were similarly cast from one pattern and differently machined. Differences between upper and lower members include: vertical depth; the omission of curves on the underside of the upper framing pieces (A and C, 163); and the location of bearing plate holes.

The depth of the upper frame is shown as less than that of the lower frame and it was thought at one stage that the upper frame would need to be strengthened. The idea of strengthening the upper frame arose from attempts to separate and apportion the load of the calculating mechanism between the upper and lower frames. For example, assigning to the upper frame, via the counter balancing springs, the weight of the warning shaft assemblies, sector and figure wheel shafts, figure wheel shafts and zeroing arms; assigning to the lower frame, via the figure wheel supports, the weight of the figure wheels. Separating the loads in this way suggests that the upper frame is more heavily loaded than the lower frame and that the aesthetic of a more slender upper frame conflicts with the load distribution. However, these considerations ignore the structural role of the figure wheel supports which couple to two frame assemblies and stiffen the whole structure. The twenty four figure wheel supports (three per axis), fixed between the upper and lower bearing plates, act as pillars or struts giving rigidity to the structure. The overall length of the figure wheel supports was tied to close tolerances to aid uniform distribution of load between the two frames. Measures to strengthen the upper frame were subsequently abandoned and the original dimensions were adhered to.

*Verticals*

The uprights were extended to the full height of the engine to allow the upper tray to be let in and still retain the overall height for the calculating axes. The frame ends have three fixings to the verticals at each corner. The two long frame members screw to the frame ends and are lapped. Fixing is by tapped recessed cheese head screws.

*Rack Mounting*

Drawing References:

| | |
|---|---|
| BAB [A] 160 | [Untitled] |
| BAB [A] 161 | General Plan and Detail of the Calculating Part (See Elevation "L") |
| BAB [A] 163 | Elevation of Difference Engine No. 2 (1/4 size) |
| BAB [A] 171 | Difference Engine No.2 Addition Carriage and mode of Driving the Axes |

Sets of toothed racks meshing with toothed quadrants (171) provide the reciprocating rotary motion for the odd and even figure wheel axes, the odd and even sector axes and the odd and even warning resets. The racks are mounted on sliders which run in channels machined in the rack mountings. The sliders are driven by links to the pivoted cam followers. The linear reciprocating motion of the racks is produced from the rotating cams by the arrangement of cam followers, links, and sliders.

Details of the rack mountings are well defined though details are not immediately obvious from the two dense drawings 160 and 161. The racks for the odd and even figure wheel axes are mounted in a single machined casting (160). The rack mounting for the odd and even warning reset axes is similarly one machined casting. However, the odd and even sector axes have separate rack mountings. There are therefore four separate cast rack mountings in all.

The figure wheel axes and the warning axes are in line so the first odd warning axis (seventh difference, extreme right) is masked in 163. However, 163 shows the bevel drive for the first odd carry shaft indicating the inclusion of a carry mechanism here. The intention to include a full carry mechanism for the seventh difference is confirmed by 161 and 164.

Nonetheless, the carry mechanism for this axis was considered to be redundant. The warning and carry axes were therefore omitted. The half–boss encircling the redundant warning axis was also omitted from the figure wheel rack mounting (F531). The profile of the curve for the right hand end of the figure wheel rack mounting was taken from elevation 163.

The figure wheel rack mounting is shown fixed at the base only. Three additional mounting ears were added to the top of the casting to provide additional back–to–front support. These secure the figure wheel rack mounting to the odd/even rack mounting (F331). The additional support was provided as a precaution against strain from side thrust in the event of jams or even resistance during normal use.

*Bearing Plates* (F371,2,3)

The bearing plates straddle the front and rear members of the horizontal frames and support the figure wheel axes, sector wheel axes, carry and warning axes, locks, and zero stops.

The relative positions of the addition and carry mechanism components are shown on 164 and 171. The detail on 171 shows the carry axis handed to the right. Plan 164 shows all carry axes i.e odd and even axes handed to the right. The mirroring correction requires the odd numbered axes to be opposite handed i.e with the carry axes handed to the left. Plans 171 and 176 show the positions of figure wheel supports, locks and zero stops for a consistently right–handed arrangement. The mounting holes for the locks, figure wheel zero stops, figure wheel supports, carry axis and sector wheel zero stops were drawn to the modified scheme required by the mirroring correction. Plan 176 shows odd and even bearing plates identical. The shape of the odd axes bearing plates have been altered to accommodate the back–to–back arrangement of the zero stops. The position of the locks is unchanged. The upshot is that the odd and even bearing plates are handed differently.

The upper and lower bearing plates have small but significant differences. The counter–bores are handed in the case of the fixing holes for the figure wheel supports and the sector zero stops. The counter–bores for the bearing plate fixing holes are not handed. Similarly, the shaft bearing holes for the carry axes, warning axes, sector wheel axes, and figure wheel axes are counter–bored but not handed. The pair of bearing plates for the constant difference (first odd axis) has the warning and carry axes shaft bearing holes omitted as the carry mechanism for this axis was considered to be redundant.

*Base Supports*

Drawing References:
BAB [A] 163, BAB [A] 165, BAB [A] 167
F381 A and B

The base supports consist of two long cast members with a basic right angle cross section running the length of the machine front and rear. Elevation 163 shows the right hand vertical framing piece resting on a boxed platform cast into the base support. The left vertical rests on a cantilevered platform. The cam stack pillars are also shown resting on a cantilevered platform (163 and 167). The right cam stack pillars rest on cylindrical bosses on the end mounts (167, F383).

The three [A] drawings also specify three shaft mountings which support the printer drive shaft along its length. The shaft mountings straddle the two base supports. One shaft mounting is provided at each end of the base supports with the third midway along.

A fourth shaft mounting was provided just inside left of the printer drive gear to give additional support to the printer drive shaft. The three shaft mountings to the left are identical (F382). The right hand end mounting (F383) has two bosses for the cam stack pillars and two half–bossed holes for the sliding bars support (167 left).

Each of the base supports was thickened in four places by the addition of pads on the lower surface of the base supports. These are positioned at each far end and under the verticals.

The purpose of the added thickness is to cover the protrusion of the cam stack pillar bolt (167) and to provide a machined surface for levelling jacks.

*Cam Stack Support*

Elevation 163 shows the cam stack pillars secured below to the base supports. No upper fixing is shown and the drive shaft assembly provides the only other securing for the cam stack assembly. Upper framing ties were added keep the drive shaft aligned and to reduce the risk of the machine breaking its back when moved. 164 and 159 show plan views of the cam stack framing plates. The upper framing plate (F424) was extended by the addition of two ears similar to the those shown for the lower framing plate (163, 159). The upper framing ties fix the ears to the verticals to secure the upper portion of the cam stack.

*Drive Shaft Bearing* (F421)

There is an inconsistency in the dimensions given for the drive shaft bearing as shown in 159 and 163. The drive shaft bearing is drawn full size on 159 (faint, lower left) with the dimension to the shaft centre as 5.25 inches. 163 shows same dimension at 4.5 inches. The discrepancy is accounted for by reference to the manuscript annotations on 159 for the thicknesses of the framing plates, and comparing these to the thicknesses as drawn on 160. For example, the bottom cam stack framing plate is drawn 0.5 inches thick in 160; manuscript annotation on 159 indicates that the plate should be increased to 1 inch. Similarly, the upper framing plate is drawn 0.5 inches thick on 160 but the annotation on 159 calls for 0.75 inches. The middle framing plate is drawn and specified at 0.5 inches. The discrepancy between the plate thicknesses as annotated and the thicknesses as drawn is 0.75 inches which is the difference between the distances to the shaft centre as shown variously on 159 and 163. The annotated thicknesses were used in preference to the drawn thicknesses and the distance to the shaft centre on the drive shaft bearing was specified at 4.5 inches (F421). The plate thicknesses appear to have been amended by annotation on 159 without amending the shaft bearing (159) or the cam stack layout (160) which had already been drawn. 163 is drawn to the annotated dimensions and was used as the basis of the layout for the machine.

## Build

The following section identifies issues of construction, assembly, and setting up that arose during the engine build.

*Base Rails*

Cast base rails were delivered dog–legged at the printer end i.e splayed outwards. We were contractually entitled to replacements but pressure to complete discouraged recourse to remaking. The alignment deficiencies in the castings were not corrected by machining prior to supply. The fixing holes for the upright framing members and for the legs supporting the bearings (which straddle the rails) are drilled into the machined palms on the rails. These were not aligned along the length of the rails. It appears that they were marked out by reference to two inconsistent datums possibly resulting from two separate stages of planing. The problem was solved by elongating the fixing holes in the legs straddling the rails and in the base of the uprights.

*Cam Profiles*

The cam profiles are determined for the ideal case in which the locus of the follower is a rectilinear path along the radial line of the cam i.e. the path that would occur with follower arms of theoretically infinite length. However, conjugate cam pairs have finite cam follower arms (of equal length) which rotate on the same pivot i.e. the locus of the followers is an arc, not a line, with a radius equal to the length of the follower arm. For the idealised case, the rises on one cam would be mirrored as symmetrical inversions for the falls on the conjugate cam. In the actual case of short pivoted follower arms, cams designed for ideal follower trajectories could result in clearance gaps between the followers and the cams, or interference.

For example, an anti–clockwise rotating cam with a rise on the main cam drives the follower outwards (170 helps to visualise this case). If the arm is trailing then the sweep of the arc has the effect of lagging the follower behind the position it would take were the locus a linear radial one. At a given point in the rise the outward displacement will be slightly less than ideal with the worst case deviation from the radial line occurring at mid–rise. The effect of the trailing follower on the conjugate follower is to displace it towards the fall by less than would be the case for an ideal follower arm of infinite length. In this case the effect of the lag of the main follower on the conjugate follower is to create a clearance between the conjugate follower and the fall. However, there is a self–cancelling effect. The follower arm on the conjugate cam is a leading arm. Because of the finite radius of the follower arm, the arc of the follower locus reaches forward, as it were, into the fall i.e. the follower leads the position is would occupy were the geometry ideal. The leads and lags so produced are self–cancelling.

In the example cited, the rise of the main cam is taken as the drive. However, in the general case of the cam design it cannot be assumed that all rises, whether on the main cam its conjugate, provide active drive: in the case of vertical motions the fall could be active in lowering the weight of an axis under gravity and/or controlling a return motion biassed from the neutral position by a counterbalancing spring. So, similarly, if a trailing follower was faced with a fall it would lag and therefore not clear the fall as quickly as an ideal follower. The conjugate leading follower would reach into the rise and be displaced outwards earlier than the ideal. Again the tendency of the lag and lead to create either clearance or interference respectively is self–cancelling. The general direction of deviation from ideal is as follows: if the main cam is leading then the non–ideal locus will act to produce interference; on the conjugate cam the deviation from ideal on trailing follower will act to produce a clearance; a leading follower on the main cam in the face of a fall will produce clearance; and the conjugate trailing follower on the rise, interference.

Since the effects of the non–ideal follower trajectories appear to be self–cancelling no account was taken of deviations from ideal loci when the cam profiles were specified i.e. the cam pairs were specified with symmetrical rises and falls. Since the vertical motions of the axes are comparatively small the undulations in the profiles of the corresponding cams are proportionately modest. In contrast, the circular motions cams show more extreme variation. The effect of the non–ideal behaviour of the followers was therefore ignored in the case of the vertical motion cams and these were cut, finished and hardened without correction. However, in the case of the circular motion cams, there remained some uncertainty as to the correctness of the reasoning that the effects were self–cancelling. Since the cam profiles are critical to the correct functioning of the engine the circular motion cams were finalised in a two–stage process during the build.

All the rises on the six circular motion cams were cut but none of the falls i.e excess material was left in the sector corresponding to the falls. The pair was assembled on the stack and an inked roller substituted for the roller follower on the partially cut cam. When driven, the follower of the fully finished cam tracked normally, and the inked roller, driven by the rises of the conjugate cam, traced out the required profiles of the falls on the unfinished cam. The traced profile was then compared with the profile based on idealised rises and falls. The process was carried out on all the circular motion cam pairs. In each case the difference between the inked profile and ideal profile was negligible, and the conjugate cams were cut, finished and hardened with no modification. This two–stage process specifying the cams served to confirm the reasoning indicating that the effects of non–ideal follower trajectories were self–cancelling. Comparing the profiles generated *in situ* by the rises with those specified on the drawing board served also to verify the correctness of the specification before the commitment to final manufacture.

In the design of the cams the start and stop positions of the rises and falls are critical to the timing of the cycle. The effect of the non–ideal loci of the followers does not affect this timing: each circular motion follower describes a single motion which starts and finishes at either the minimum or maximum displacement, and it is clear from the geometry of the original design that the minimum and maximum follower displacements both occur on the radial line through the cam centre i.e. follower is tangential at the mean diameter. Geometrically this corresponds to the pivot point of the follower arms being chosen such that the line bisecting the sector of motion of a follower arm is at right angle to the radial line through the cam centre. The effect of the non–ideal loci of the followers is therefore to slightly alter the internal timing of the excursions of the followers *within* a rise or fall but not to alter the cam angle or the timing widow within which the excursion is completed. Since the rises and falls are arcs of circles i.e. monotone increasing or decreasing curves with no inflection points within a single excursion, there are no internal events within the timing window that might be affected by the deviation from ideal of the follower loci.

*Speed of Operation*

When the machine was first assembled the manual drive was relatively stiff and for reasons of caution the operating speeds were kept low. After several thousand cycles the manual drive was found to be noticeably freer and the run–in speed of operation is 10 complete engine cycles per minute i.e. one tabular calculation every six seconds (equivalent to 40 turns of the handle per minute with the 4:1 reduction gear operative). If the machine is run faster than this the detent wheel of the intermittent circular motion drive of the carry axes tends to overshoot and cause jamming of the drive. If the motion is erratic or slower the sectors when lowered tend not to mesh cleanly with the figure wheels and jams occur. This is thought to be related to the effects the bell–cranks flexing under load at slow speeds and producing timing lags. With sufficient uniform momentum the meshing is unproblematic in normal operation.

*Keyways*

Since the lead–in and lead–out timings are critical and the relative phasing of the cams can only be verified during the build, the keyways in cams and the main cam drive shaft were cut narrower than indicated in the drawings. This was to allow for fine timing adjustments during the build should this prove necessary. This measure allowed some leeway to fine tune the phasing between cam pairs as well as between one cam and its conjugate. In the event no adjustment was needed and the keyways were opened out to the specified size.

*Figure Wheel Drive Arms*

The figure wheel drive arms for the Trial Piece were manufactured as indicated in the original drawings i.e. with no chamfers on the undersides to provide lead–in to the internal nibs when lowered. On the Trial Piece the drive arms did not foul the internal nibs and there was therefore no advance warning that fouling might be a problem. However, with multiple stacks on the engine, lowering the figure wheel axes produced consistent jamming especially with figure wheels set at '0' or '9' at which points the clearances are small. The solution was to chamfer the undersides of the drive arms to make the engagement less critical.

**Checking Carries after Assembly** (Red Book, page106–7)

*Setting Up Initial Values*

The setting up procedure consists of a series of operations that allow the figure wheels to be set manually with the initial values of differences and the tabular value from which the calculation is to commence. The design of the calculating section features several measures to protect the integrity of the calculation: the figure wheel and sector wheel locks as well as the lugs on the carry levers restrict size of the time windows in which the figure and sector wheels are free to move as well as restricting the origin of their motions to legitimate sources only. These measures are intended to prevent derangement of the wheels during normal operation i.e. movement imparted by extraneous sources including deliberate or inadvertent manual input. These security measures act against readily inputting initial values by manually altering figure wheel settings, and the security devices need to be disabled or bypassed to allow setting up. The setting up procedure specifies the sequence of operations to be followed to allow the initial values to be set on the figure wheels.

The original design drawings are not accompanied by any contemporary explanatory text or discussion. In a very rare exception a brief textual account is given for the setting up procedure as a preface to the timing diagram (BAB [F] 385/1a) dated March 1848. The setting up sequence described takes the engine through two complete calculating cycles. The first cycle leaves the sectors disengaged, the figure wheels zeroised and residual carries cleared; the initial values are entered during the second full cycle of the set up sequence.

The procedure described by Babbage has a major flaw (see Paragraph 2 below) which became evident when the procedure was attempted in practice. The italicised sections below are transcriptions of the original text. Paragraph numbers are not original and have been added for ease of reference.

1. *The Engine is to be stopped at the end of a cycle of 50 when both sets of sectors **S** will be at zero, the bars $^{23}$**E** $^{24}$**E** which give them vertical motion ungeared by means of the axis $^{2}$**F** (see drawing 159) and all those sectors raised to their highest position by the handles $^{5}$**E** drawing 163.*

The first instruction is to disengage the sector wheels from the figure wheels. The timing diagram which follows the manuscript description (3851/a) confirms that both odd and even sectors are at zero at the end of the second half–cycle. This is to prevent the new values set on the odd axes being given off to the even axes when the engine is advanced during the set up procedure. Both sets of sector wheels are therefore fully disengaged (raised to their highest point) for the duration of the set up procedure. The sectors are disengaged by operating the two lifting handles (163 **E**, **E** [Notation Refs?], detail 177). The left hand handle lifts the odd sector axes to the fully raised position; the right hand handle raises the even sector axes.

At the end of the second half–cycle (*'the end of a cycle of 50'*) the odd sectors are already disengaged (fully raised) and the left hand lifting handle can be locked in place with the pull–out plunger. However, to raise the even sectors, the horizontal sector bars ($^{23}\mathbf{E}$, $^{24}\mathbf{E}$) need to be uncoupled from the horizontal drive at the cam stack end. This is achieved by lifting the release lever ($^{2}\mathbf{F}$, 159 centre and 168 top right) which lifts both the odd and even bar levers out of their drive slots in the sector bars. This frees the lifting handle to move the sector bar. The lifting handle is locked in the raised position as before with the pull–out plunger. Disengaging the sector bars prevents the vertical motion drive from conflicting with the now immobilised sector axes when the engine is cycled through the set up procedure. (The lifting handles waggle slightly during normal operation.)

2. *Reduce all the Fig. wheels $^{1}$ [Delta] to zero by moving the driving axis $^{1}\mathbf{A}$ once round.*

This step contains a major flaw evidently overlooked in the original procedure which leads to the setting up process being self–corrupting. This arises through a process in which the figure wheels tend to be dragged back from their zero positions by residual friction with the figure wheel axes during the return stroke of the axes while the figure wheels are neither locked, nor meshed with the sector wheels. To expand: reference to the timing diagram shows that at the start of the cycle the odd figure wheel locks disengage, the odd the figure wheel axes at the same time lower to bring the drive arms into the plane of the internal nibs, and the odd figure wheels are then driven to zero and then locked. The odd figure wheel axes then lift to disengage the drive arms. During this process the odd figure wheels remain engaged with the sectors which then restore the figure wheel number just given off. So during the first half cycle of normal operation the odd figure wheels are being driven by the drive arms, are locked, or being restored by the sectors. However, the first step of the set up procedure was to disengage the sectors. So during the set up procedure (as distinct from normal operation) when the locks free the figure wheels for restoration by the sectors after giving off, the figure wheels are not secured and it was found that the return of the figure wheel axes, after zeroising dragged back some of the figure wheels a varying amount depending on the differing amounts of friction. The intended zeroising action is thus subverted. Moreover, when the locks attempt to re–engage at the end of the first half–cycle jams occurred on the wheels that were deranged by other than a full digit interval. In normal operation any drag of this kind would be unlikely to derange a figure wheel meshed with a sector wheel because of the additional load. But even if it did, the direction of motion simply anticipates the action of the sector wheels to restore the figure wheel values and any drag–induced motion is non–corrupting. (During normal operation the restoration of the figure wheel values occurs during the same period as the return stroke of the figure wheel axes). The process of zeroising the even figure wheels during the second half cycle will be similarly thwarted by variable derangement. Whether or not figure wheels are deranged so as to foul the locks is unpredictable. Even if jamming the locks was not a problem the same dragging action of the figure wheel axes would corrupt the initial settings during the second cycle of the setting up procedure. This flaw renders the original set up procedure ineffective and could not be left unremedied.

The solution was to provide manual locks for each of the axes. These consist of vertical lengths of steel fixed to the front–most figure wheel supports of each of the axes. The eight setting locks are fixed to the vertical supports by knurled screws passing through slotted holes in the locks. In normal operation the locks are retracted and play no part. During setting up each lock is freed by partially unscrewing the fixings by hand. The lock is slid forward to engage with the column of figure wheel teeth and secured in the locked position by retightening the fixings. The locks come into play twice during setting up: they secure the figure wheels in the zero position during the first set up cycle and they secure the figure wheel initial settings during the second cycle (see below). The odd axes setting locks are engaged at the 10–unit point i.e. immediately after the odds axes values are given off by driving the figure wheels to zero. Engaging the locks at the 10–unit point ensures that the figure wheels can be zeroised by the figure wheel axes drive arms without obstruction. The even setting locks are engaged at the 35–unit point for the same reason. The revised set up sequence which includes the operation of the setting locks is given after step 5 below.

3. *Move the axis '**A** 20 units more , and set the Odd Difference Figure wheels '.*

This is the start of the second full cycle in the set up procedure. The 20–unit point occurs during the period of normal operation in which the figure wheels would be restored by the sectors. However, the sectors have been manually disengaged and the locks withdrawn, so the figure wheels can be turned freely by hand to the required initial values. The reason for advancing the engine to the 20–unit point is that this is the point in the cycle at which the carry mechanism is inactive i.e. the odd carry levers have been driven against the reset stop by the sweeping motion of the detent support arm. The figure wheels can be turned in either direction without regard for producing spurious warnings i.e. the wheel can be turned to any of the four decades with the confidence of knowing that the carry mechanism will not engage. The choice of this point in the cycle is an elegance. Warnings set by manual set up prior to the 17–unit point would be cleared automatically; those set after the 20–unit point would remain as residual warnings and would have to be cleared by hand. The elegance of performing manual settings at the 20–unit point is twofold. Firstly. the there are no spurious warnings to reset and therefore no danger of residual warnings from the setting up procedure corrupting the calculation. Secondly, the figure wheels can be rotated in either direction: if the carry mechanism was operative the curved warning limb of the carry lever would foul the carry finger on the figure wheel if rotated in the wrong direction.

4. *Move the axis* $^{1}\mathbf{A}$ *25 units more, that is o the end of the 45th unit, and set the even Diff. Fig. wheels* $^{1}$.

The 45–unit point is the even–axis equivalent of the 20–unit point for the odds axis and the same considerations (see 3 above) apply with respect to the elegance of choosing this part of the cycle for setting initial values.

5. *Move 5 units more, that is to the end of the Cycle of 50, gear the sectors by means of the handles* $^{5}\mathbf{E}$ *and axis* $^{2}\mathbf{F}$ *and all is ready for commencing the calculations.*

This step re–engages the sectors by releasing the lifting handles and re–engages the drive arms with the sector bars to restore the drive from the cam followers.

*Revised Setup Procedure*

The revised sequence of operations for setting up taking account of the setting up locks introduced to overcome the self–corrupting effect of the unmodified procedure is given below. Reference to units of the cycle refer to Babbage's 50–unit cycle which corresponds to one complete rotation of the main drive shaft and one complete calculation. The 50–units of the cycle are engraved on the chapter wheel which rotates with the main drive shaft and the position in the cycle is indicated by the pointer fixed to the vertical framing member in full view of the operator turning the handle.

*Setting up Procedure*

1. Cycle the engine to the zero point i.e. the end of a cycle.

2. Lift the release lever at the cam stack to disengage the drive to the sector bars.

3. Rotate in turn each of the two lifting handles anti–clockwise to fully disengage the sectors (the movement is only a few degrees).

   When each of the handles reaches the end of its travel, lock it in the raised position pulling the locking plunger fully out so as to secure it with the bayonet locking device. (It is important that the sectors are raised to the highest point to avoid fouling the figure wheels during setting up. The way to ensure this is to check that the locking plungers are slid fully home towards the operator standing in front of the engine).

4. Advance the engine to the 10–unit point.

5. Engage the odd setting locks. (Unscrew the knurled fixing screws, slide the locks into engagement, retighten the fixings).

6. Advance the engine to the 35–unit point.

7. Engage the even setting locks (see step 5).

8. Complete the first cycle and advance the engine to the 20–unit point ('SET ODD' on the chapter wheel).

9. Disengage the odd setting lock on the axis being set up. (Unlocking only the axis being set up avoids inadvertent derangement of axes already set up).

10. Enter the odd difference values on the figure wheels by rotating each figure wheel by hand to read the required value. The figure wheel value is indicated by a cursor/arrow engraved on the figure wheel supports above each figure wheel. The least significant digit position is at the bottom, the most significant digit position is at the top.

*Automatic cycle counting*

If a section of the tabular column (last even axis) is to be used as a cycle counter then when the first difference column is set up a '1' should be placed on the figure wheel opposite the least significant position of the counter. Each time a calculation cycle is performed the counter will be incremented by 1 and the counter will register the cumulative total number of calculations performed. An additional elegance is to enter the start value of the independent variable in the section of the tabular column reserved for the cycle counter and enter a '1' in the first difference column in the counter's least significant digit position. The value of the variable will be incremented by '1' each calculation cycle and the counter will directly keep track of the value of the variable.

11. Re–engage the odd setting lock.

12. Repeat steps 9, 10 and 11 for each odd axis required for the calculation.

13. Advance the cycle to the 45–unit point ('SET EVEN' on the chapter wheel).

14. Disengage the even setting lock on the axis being set up.

15. Enter the even difference values on the figure wheels by rotating each figure wheel by hand to read the required value. The figure wheel value is indicated by a cursor/arrow engraved on the figure wheel supports above each figure wheel. The least significant digit position is at the bottom, the most significant digit position is at the top.

16. Re–engage the even setting lock.

17. Repeat steps 14, 15 and 16 for each even axis required for the calculation.

18. Advance the cycle to the zero–unit point i.e. the end of cycle.

19. Retract the locking plungers to release the two sector lifting levers and re–engage the sectors. Ensure that the locking plungers are fully retracted and locked in this position by the bayonet fixing.

20. Re–engage the sector drive by lowering the release lever at the cam stack. The drive

levers may require jiggling to allow them to drop into the slots in the sector bars.

21. Disengage all the setting locks and re–tighten the knurled fixing screws.

It is imperative not to deviate from the fixed recommended sequence of the procedure and any temptation to take short cuts by omitting any of the steps should be resisted. It is especially important to strictly observe the sequence for engaging and disengaging the setting locks. If a setting lock is left engaged and a residual warnings left uncleared by omitting part of the procedure then the figure wheels are immobilised and the steel carry arms will snap the bronze carry levers during the carry cycle.

Babbage's rare and welcome lapse into text [385/1a] in which he describes the setting up initial values calls for two complete cycles for the procedure. It is not evident that the first cycle is indeed indispensable to the process. The first cycle leaves all figure wheels zeroised and spotting non–zero values is a convenient visual check of the correct operation of giving off and carry clearance. Apart from this there is no evident reason why the cycle cannot be omitted. There may in fact be some advantage in dispensing with it: when the odd axes are set up (20–unit point) these will be at zero while the even axes will retain their residual values. In progressing along the machine from odd axis to odd axis, the odd axes are more easily identifiable by their fully zeroed values and there it is less likely that an even axis will mistakenly be altered. This is a marginal psychological benefit that might assist in avoiding operator confusion but one which is perhaps outweighed by the value of the first cycle as a verification of the giving off and carry clearance function.

*Checking Carry Operation*

The checking procedure described should be used as part of the commissioning process after first assembly and to check a column after reassembly following removal for repair or inspection. Moreover, it serves to verify a suspect column after the machine has been in service.

The warning and carry axes are assembled on the bench and then offered up to the machine. The reset stops (the comb–columns) are fixed after the warning axes are in place. If the procedure is being used after the first assembly then the reset stops should be left off the machine until the appropriate step below. A warning axis and its corresponding carry axis are checked as each pair is installed i.e. the commissioning and verification process is progressive: each of the seven difference positions are checked in turn. The procedure described allows the verification of the operation of one difference position consisting of a thirty–one digit figure wheel column, a carry axis and warning axis. [Ref: Red Book, page 106].

*Preliminary*

1. Uncouple the sector wheels by raising them to their upper–most position. This is done using te sector uncoupling levers (front lower right horizontalframing member). Advance the engine to or 2–unit point on the chapter wheel if column being checked is even, or to the 27–unit point if column being checked is odd. This ensures that the warning axis is in the raised position which is necessary for the first phase of checks i.e verifying the warning mechanism, and unlocks the figure wheels.

*Warning Check*

2. By hand set all bronze carry levers in the column to the "unwarned" position (first notch on the detent lever).

3. Check that each of the four outer nibs on a given figure wheel advances the carry detent limb one notch from the "unwarned" to the "warned" position. This is best done with two operators, one at the front of the engine and one behind. Each time the mechanism is warned by the front operator rotating the figure wheel, it is restored by the rear operator to the unwarned position by hand. The direction of rotation here is critical: if the column being checked is an even figure wheel column then the figure wheel should be rotated anti–clockwise; if the column is an odd one, rotate clockwise.

4. If the carry detent limb overshoots (the v–shaped insert ends up resting on the shoulder past the detent notch) or undershoots (fails to reach the notch) then adjust the bend of the warning limb using the special tool. to eliminate any over- or under-shoot of the carry detent lever.

5. Repeat the procedure described in the previous step for each of the figure wheels in the column.

6. Restore each carry lever to the "unwarned" position.

*Carry Check*

Checking the engagement of the locking lugs on the carry lever.

7. Swing the lowermost carry lever (this has the long arm removed) out of the way i.e. so that the locking lugs are clear of the figure wheel.

8. Cycle the engine slowly to lower the warning axis: if the column is odd, advance to the 36–unit point on the chapter wheel; if the column is even then advance to the 11–unit point. While the axis lowers check that the locking lugs on the carry lever all mesh smoothly and do not derange the figure wheels. If any figure wheel motion is detected go back to the previous step and investigate. The engine may need to be cycled several times while different sections are observed.

9. Position the lowermost carry lever so that the lugs on the carry lever correctly mesh with the lowermost figure wheel and fix using grub screws.

*Checking Restoration to "Unwarned"*

10. Fix the reset stops to the figure wheel supports. This is best done with the engine at the zero unit point on the chapter wheel i.e. when the carry levers arereset to the "unwarned" position.

11. By hand set all the carry levers to the "carried" position. Check each carry lever to confirm that the carry limbs do not foul the stops and that there is asmall clearance between the lever and the stop.

12. Cycle the engine at normal speed.

13. Check that all carries are returned to the "unwarned" position. If the carry levers do ot return to the "unwarned" position then the warning rack will need adjustment.

*Checking Carry Action*

14. Set carry levers numbers one and sixteen to the "warned" position. (Lever number one is the first lever to be engaged by a steel carry arm).

15. Cycle the engine very slowly. While one operator cycles the machine the other checks that the figure wheel locks withdraw in time to allow the figurewheel to receive the carry from the decade immediately below i.e. this checks that there is sufficient clearance for the locks to accommodate a carry. This check is vital: the clearance is small when correctly adjusted and if the lock does not clear then the steel carry arm will snap the bronze carry lever. If there is insufficient clearance then the phasing of the carry axis circular motion needs to be retarded. This is done by adjusting the bevel gears on the shaft from the intermittent drive.

16. By hand set all the carry levers to the "warned" position.

17. Cycle the engine and check that all positions carry. If carries malfunction adjust the warning rack.

18. Check that having carried, all carry levers reset to "unwarned" at the end of the appropriate part of the cycle.

[Transcript [F] 385/1a]

*Notation of Units for Difference Engine No.2*
*The Drawings 147 to 177 inclusive*
*March 1848*

*The Engine is to be stopped at the end of a cycle of 50 when both sets of sectors* **S** *will be at zero, the bars* ${}^{23}$**E** ${}^{24}$**E***which give them vertical motion ungeared by means of the axis* ${}^{2}$**F** *(see drawing 159) and all those sectors raised to their highest position by the handles* ${}^{5}$**E** *drawing 163.*

*Reduce all the Fig. wheels* ${}^{1}$ *to zero by moving the driving axis* ${}^{1}$**A** *once round.*
*Move the axis* ${}^{1}$**A** *20 units more , and set the Odd Difference Figure wheels* ${}^{1}$*.*
*Move the axis* ${}^{1}$**A** *25 units more, that is o the end of the 45th unit, and set the even Diff. Fig. wheels* ${}^{1}$*.*

*Move 5 units more, that is to the end of the Cycle of 50, gear the sectors by means of the handles* ${}^{5}$**E** *and axis* ${}^{2}$**F** *and all is ready for commencing the calculations.*

***Appendix***

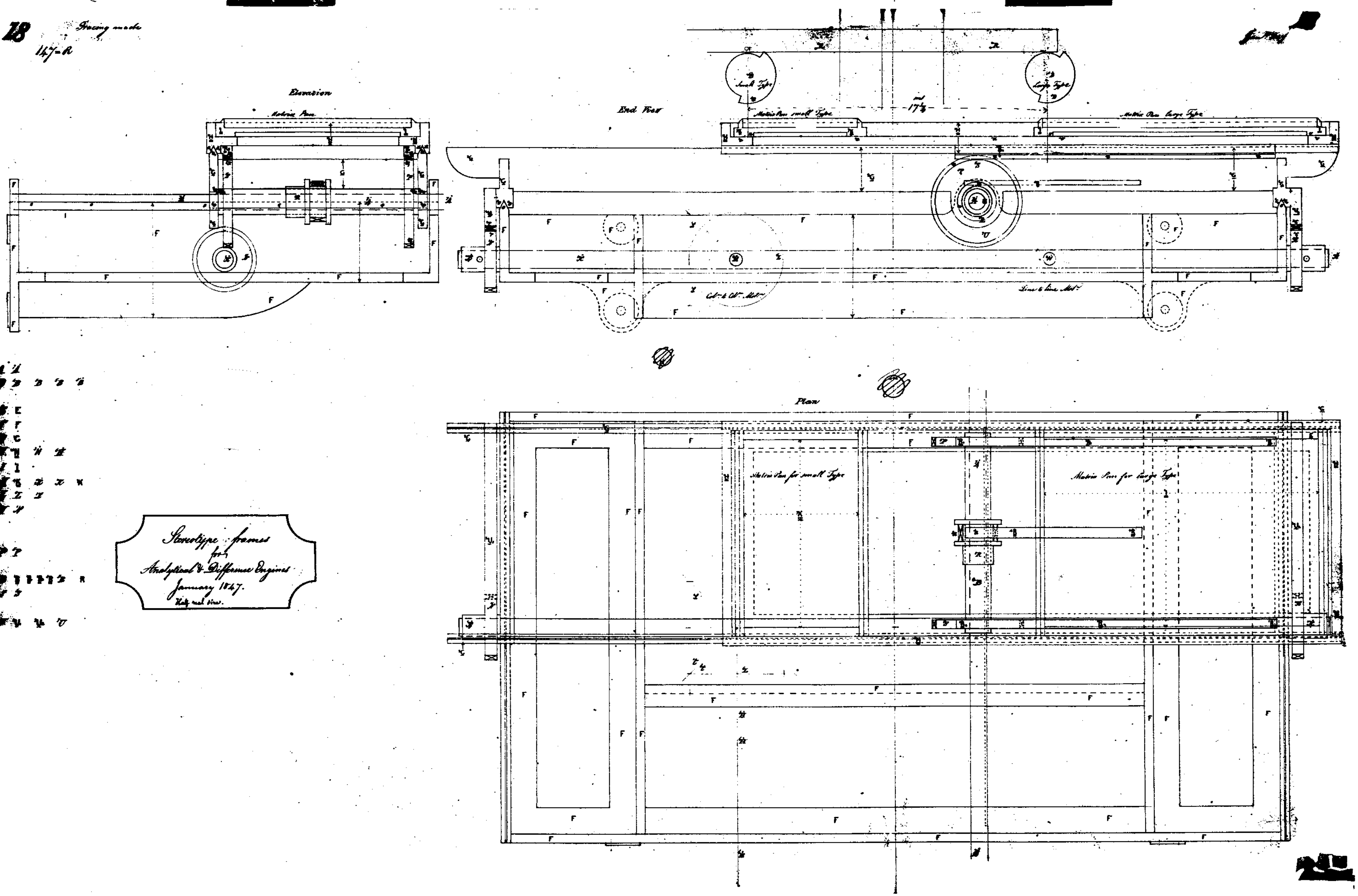
Scale: Approx. 1 inch = 14.5 inches

| | |
|---|---|
| BAB [A] 147 | Stereotype frames for Analytical and Difference Engines |
| BAB [A] 159 | General Plan and Details of Cams for Driving Calculating Axes |
| BAB [A] 160 | [Untitled] |
| BAB [A] 161 | General Plan and Detail of the Driving of the Calculating part of Difference Engine No. 2 |
| BAB [A] 162 | Part of Frame for Supporting Stereotype axes |
| BAB [A] 163 | Elevation for Difference Engine No. 2 |
| BAB [A] 164 | Plan of Difference Engine No. 2 |
| BAB [A] 165 | End View and Elevation of Paper Rollers<br>End View and Plan of of Carrying Axis and Driver<br>End View and Elevation of Supports |
| BAB [A] 166 | Apparatus for moviing Stereotype frames for Analytical and Difference Engines |
| BAB [A] 167 | Elevation and End View of Part of General Framing |
| BAB [A] 168 | Plan and End View of Cams etc for Vertical Motion to Calculating axes |
| BAB [A] 169 | Plan of Cams for Locking Odd Difference Figure Wheels and for Vertical motion of Even Difference warning |
| BAB [A] 170 | Plan of Cams for Circular Motion of Even Difference Axes |
| BAB [A] 171 | Difference Engine No. 2 Addition of Carriage and mode of Driving the Axes |
| BAB [A] 172 | End View of Inking Printing Paper and Stereotype Apparatus |
| BAB [A] 173 | Plan of Inking Printing and Stereotype Apparatus |
| BAB [A] 174 | Rack Pinions for connecting Table figure wheels with Printing Stereotype Sectors |
| BAB [A] 175 | Plan of Cams for Punching with small stereotype sectors, and Cams for removing Paper Rollers |
| BAB [A] 176 | Plan of Calculating part of Difference Enigne with the means of conveying numbers to Stereotype sectors |
| BAB [A] 177 | Difference Engine No. 2. Bars and Levers for lifting Axes |
| BAB [F] 385/1[a] | Notation of Units for Difference Engine No. 2 |
| 337 X 26 | Addition Carriage (Modified) |

Note

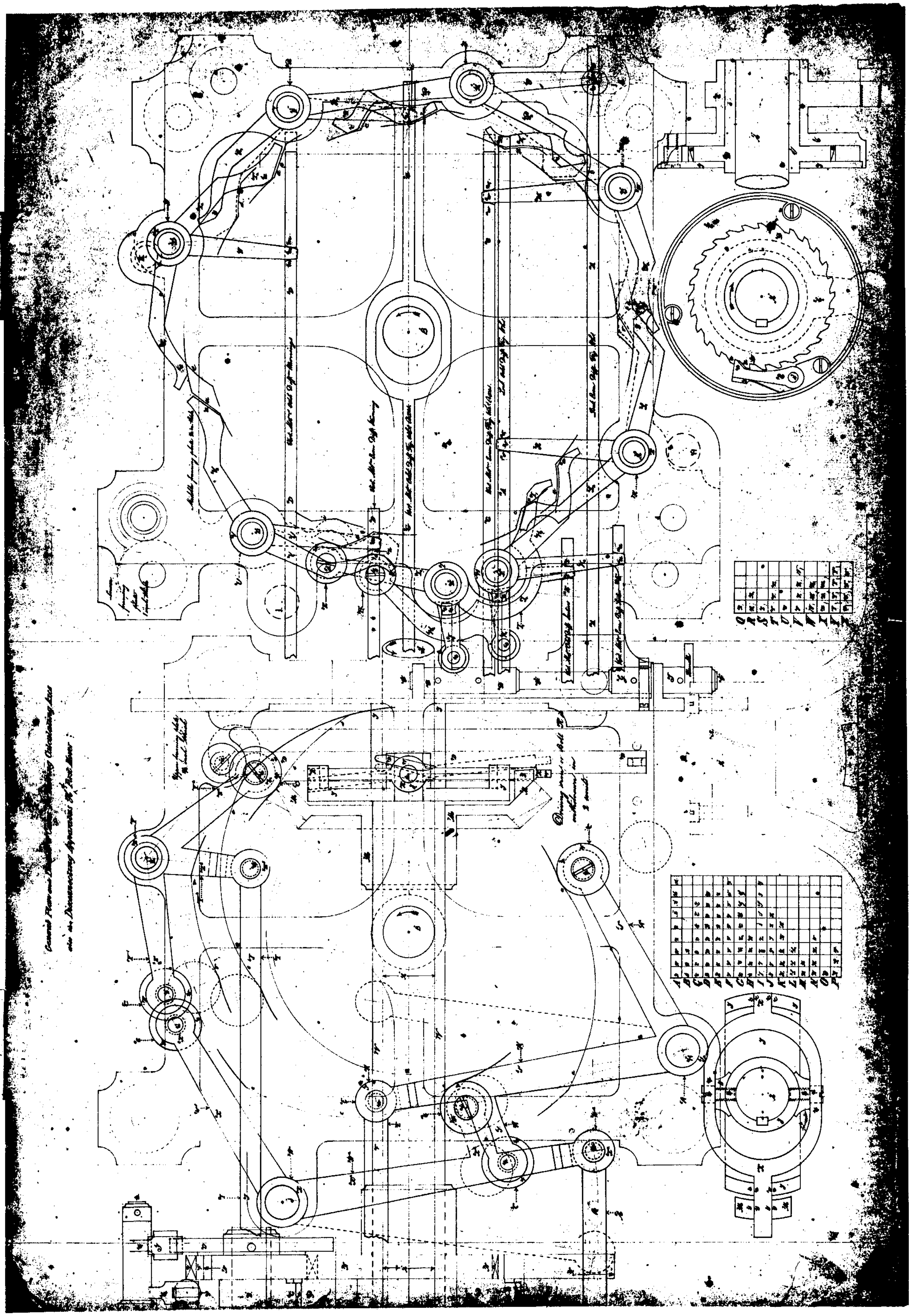
The twenty main drawings for DE2 were part of the series of drawings for the Analytical Engine identified by Babbage with the prefix [A] as above. The twenty DE2 drawings were subsequently renumbered by Babbage in the series B1 through B20. The original A-series numbers have been used here and these are as listed in the cross-referenced list compiled by Allan G. Bromley. See, Bromley, Allan G. *The Babbage papers in the Science Museum: a cross-referenced list.* London: Science Museum, 1991. 337 X 26 is a modern drawing showing the layout of the modified carriage arrangement.

28 Tracing made
147-R



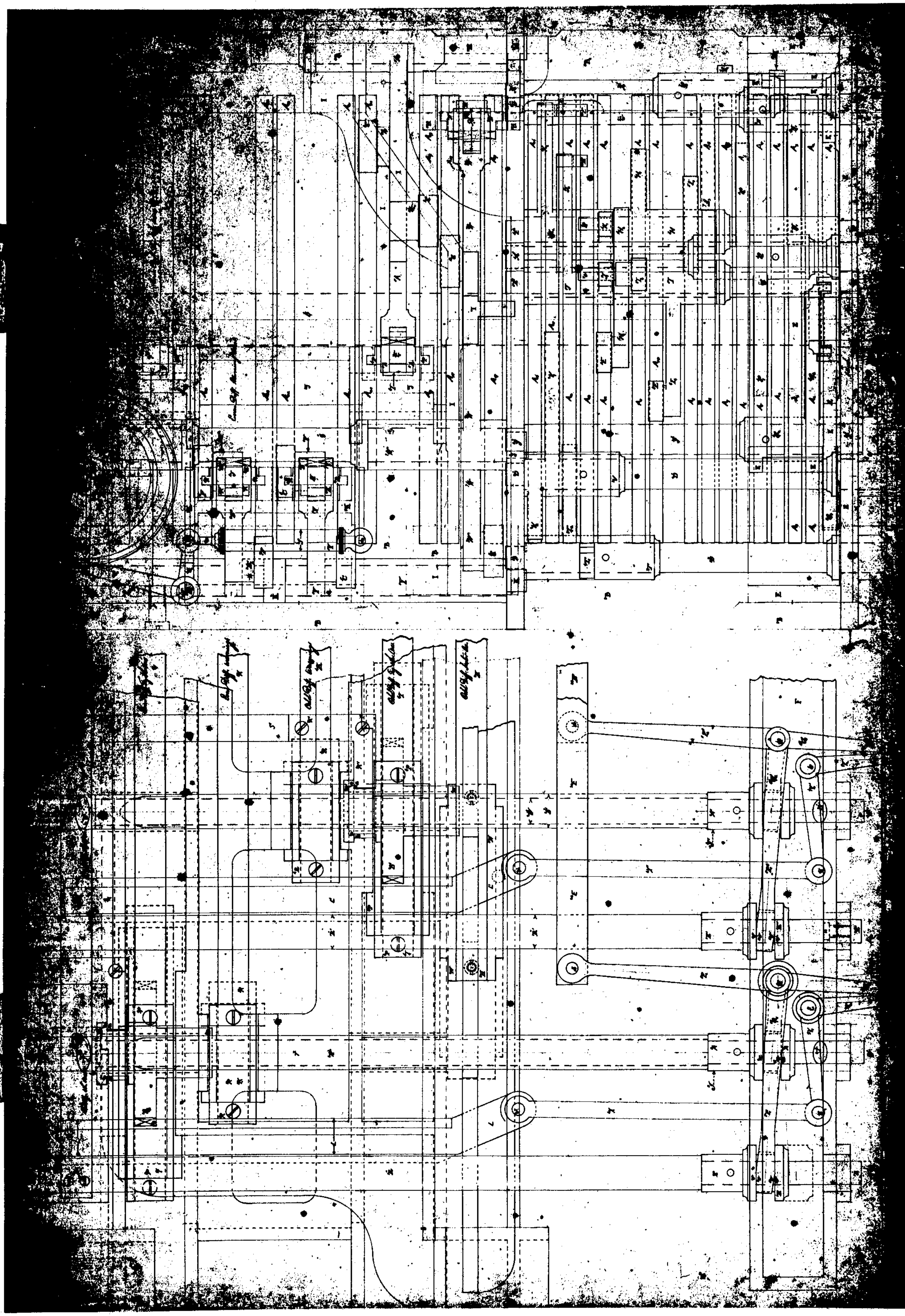

Stereotype frames
for
Analytical & Difference Engines
January 1847.
Half real size.

BAB [A] 159

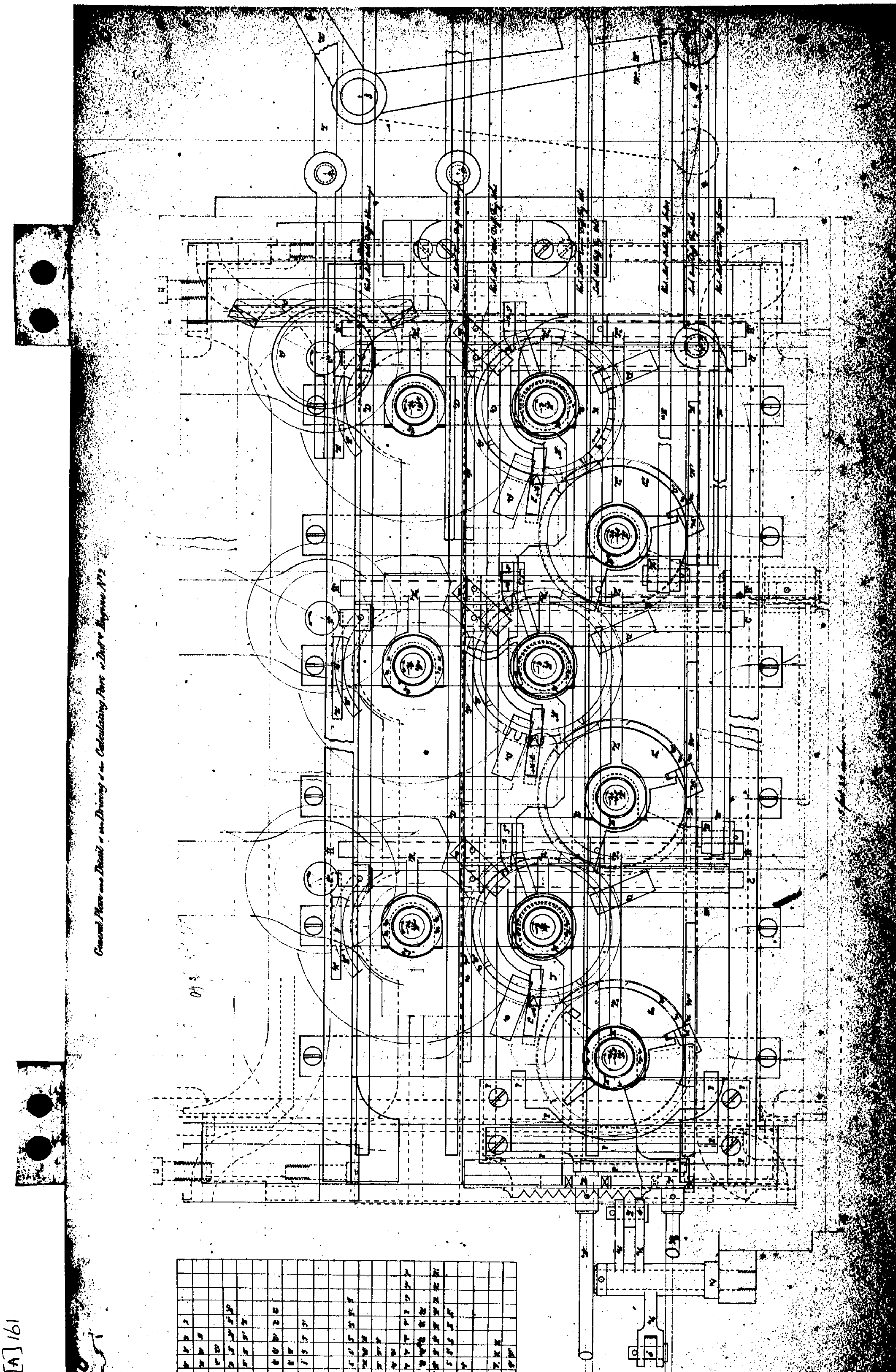


BAB [A] 159

BAB [A] 159

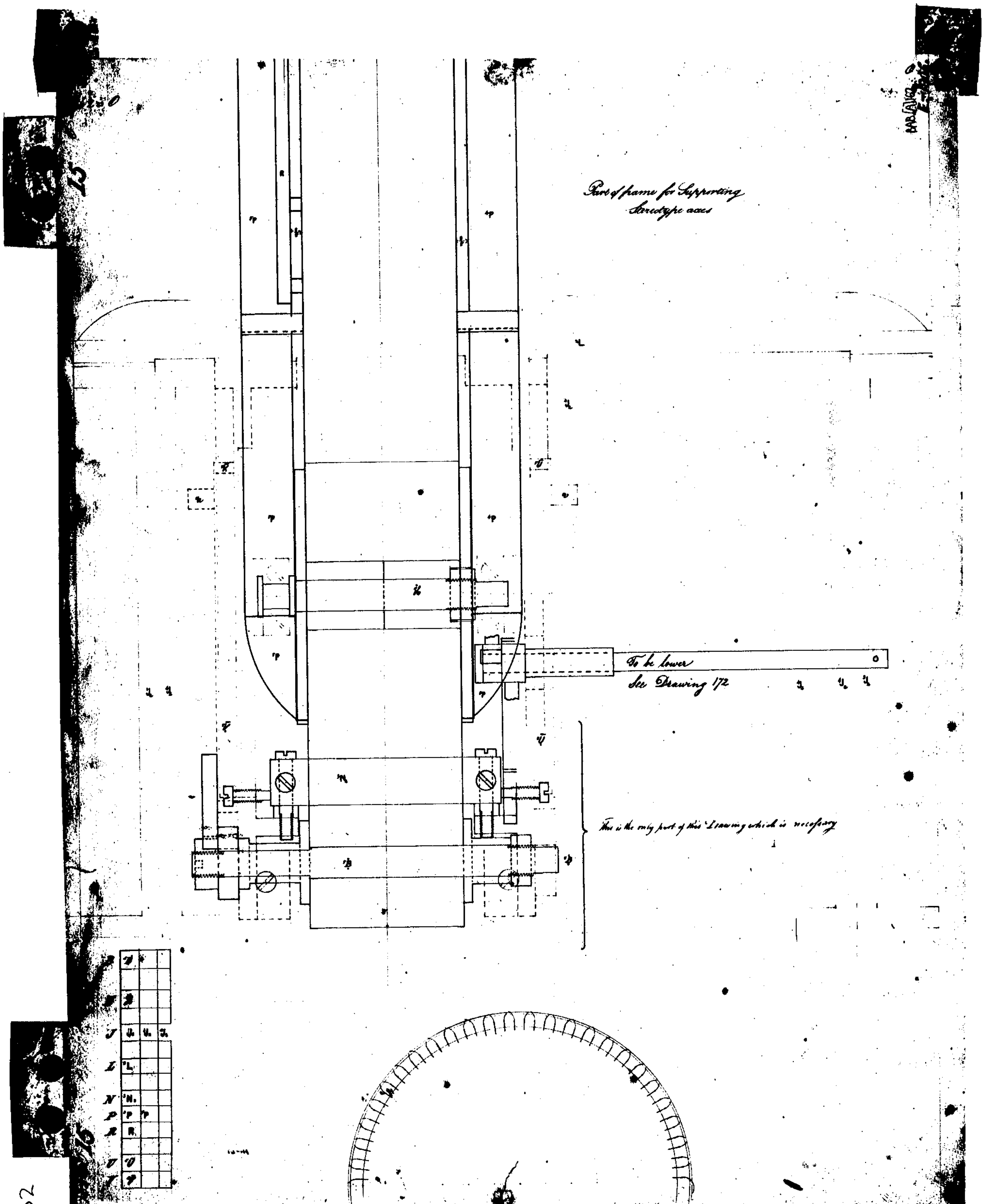


BAB [A] 160
BAB [A] 160



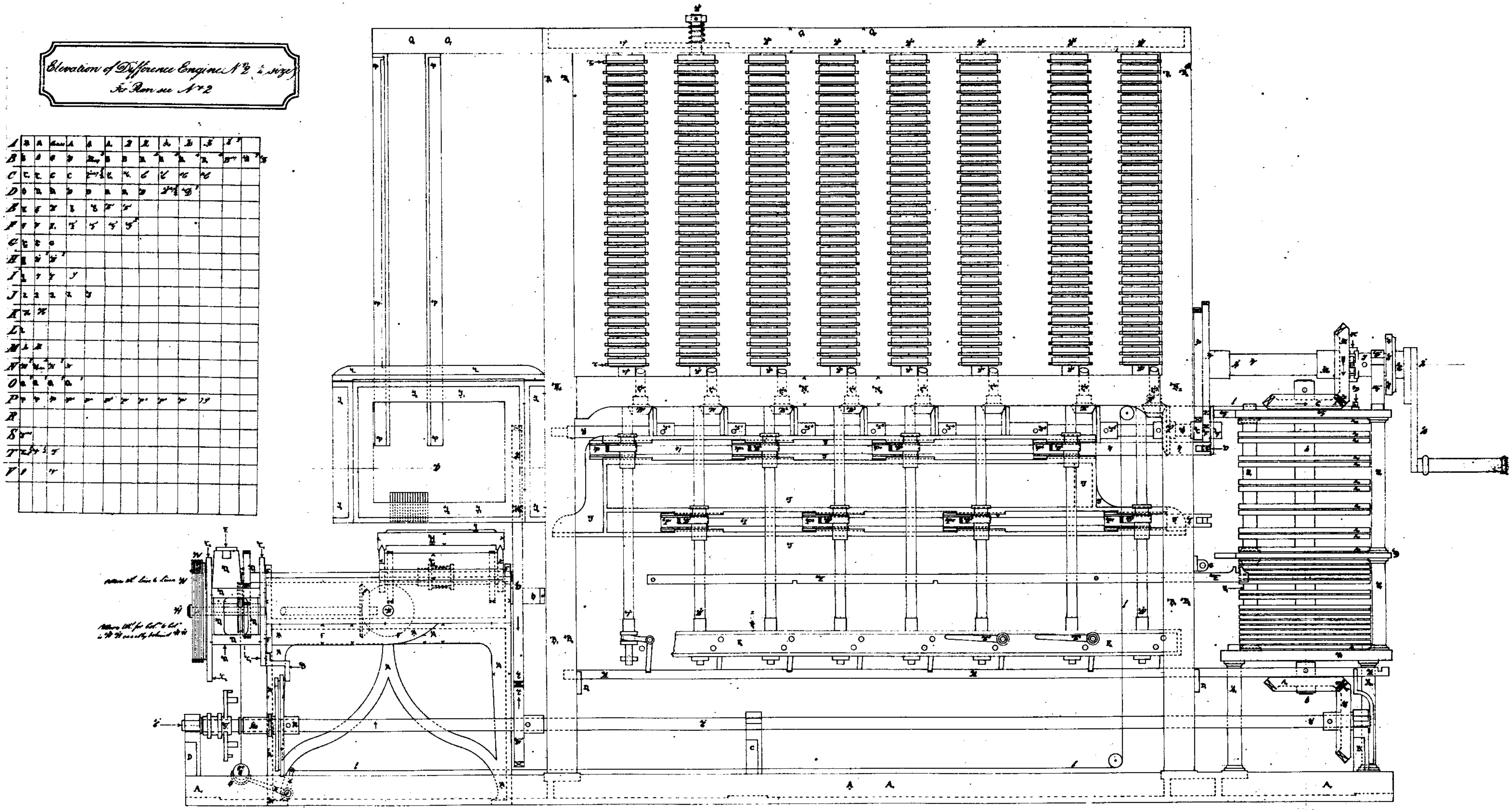
General Plan and Detail of the Driving of the Calculating Part of Diffce Engine No 2


BAB [A] 161


Part of frame for Supporting
Stereotype axes
To be lower
See Drawing 172
This is the only part of this Drawing which is necessary

3 = 6 Drawing made.

Elevation of Difference Engine No 2 (½ size)
For Plan see No 2



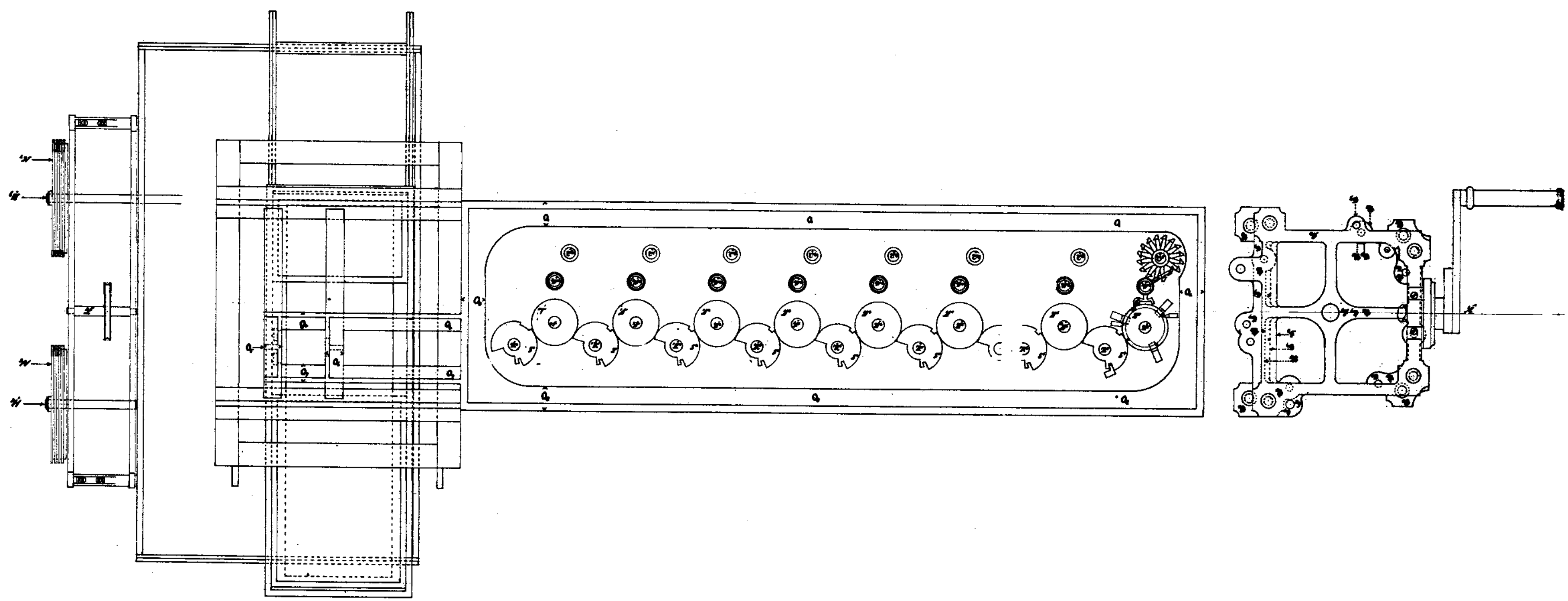
BAB [A] 163

[A] 164

Drawing Boards
164-B 2

2

Plan of Difference Engine No 2 (½ size.)

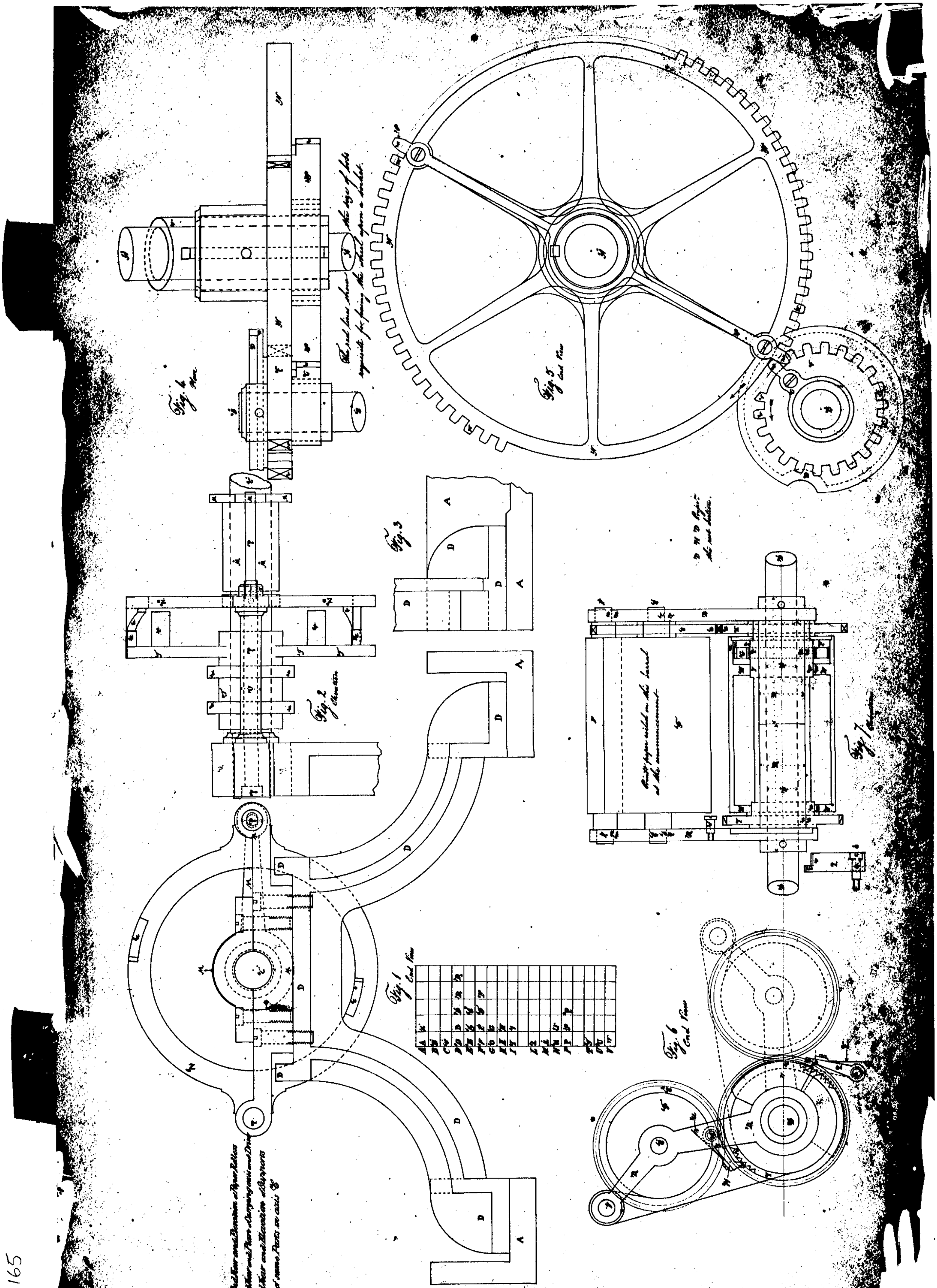




BAB [A] 164

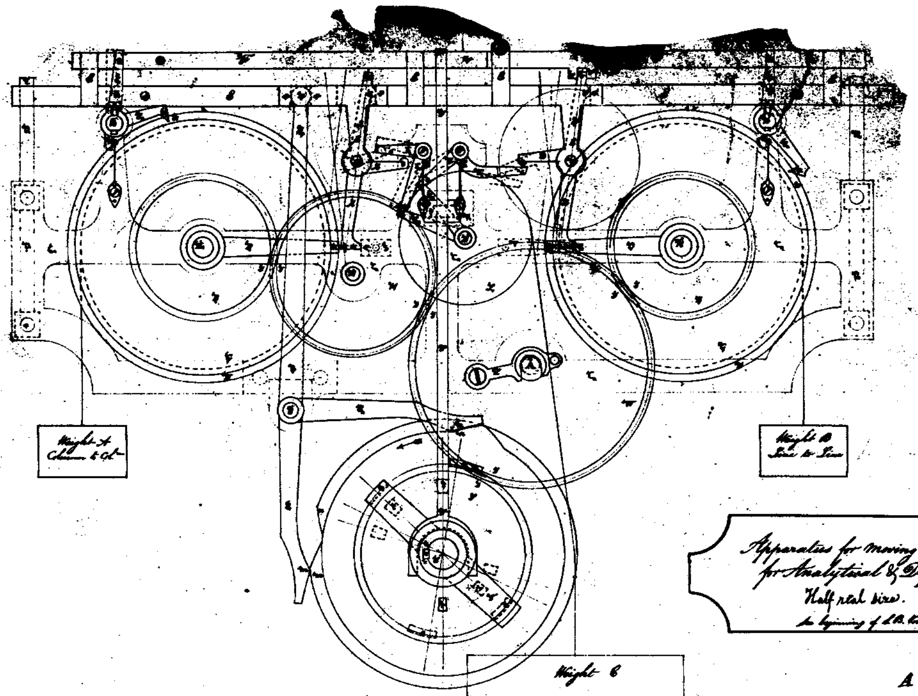
BAB [A] 165

BAB [A] 165

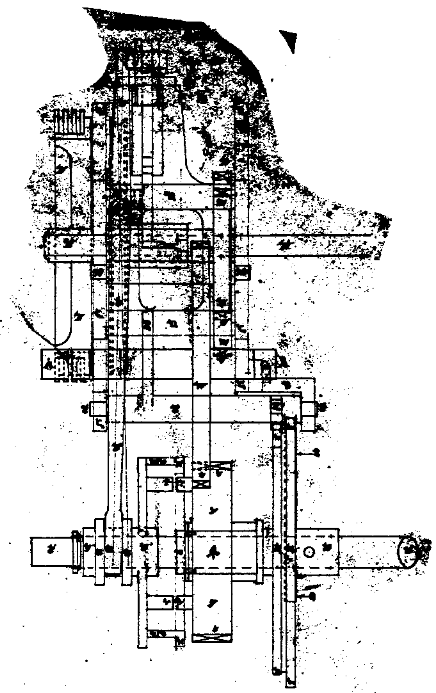



BAB [A] 165

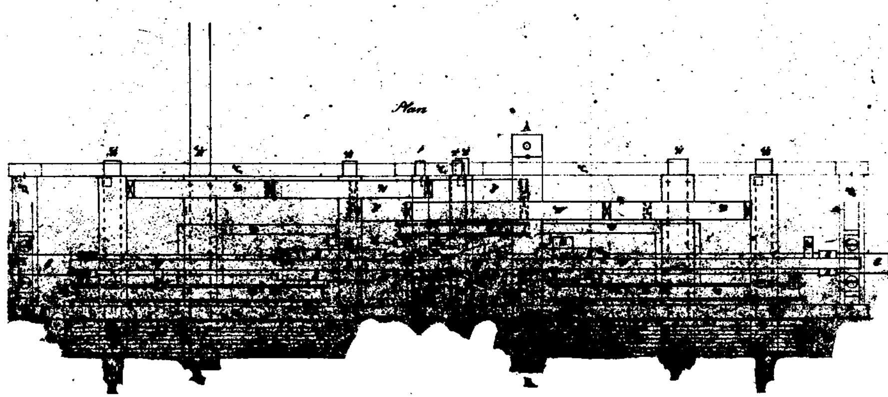
165




Apparatus for moving Stereotype frames
for Analytical & Difference Engines
Half real size.
See beginning of [illegible] page [illegible]

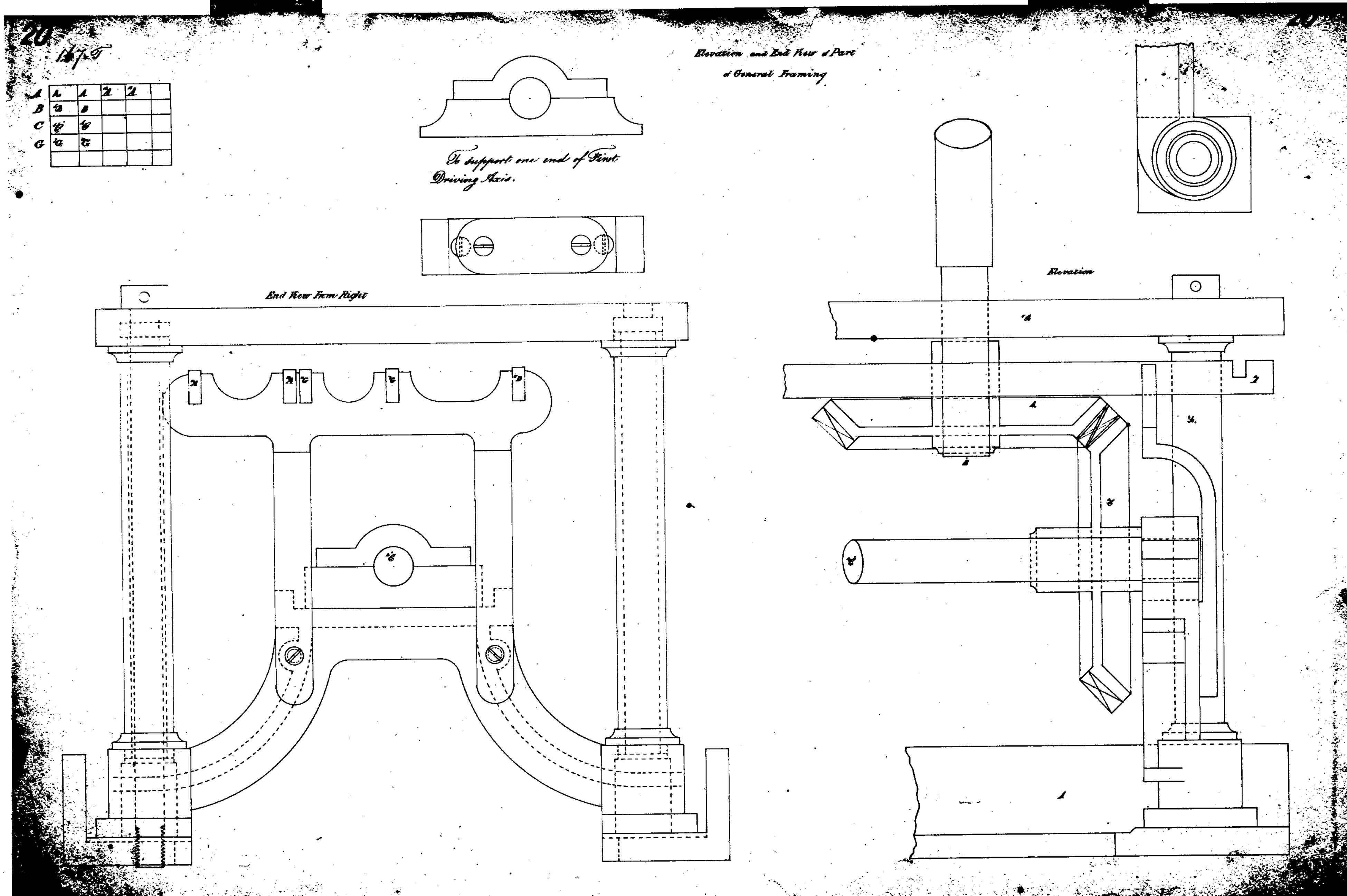


| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| A | Å, | [illegible] | [illegible] | | | | | | |
| B | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] |
| C | C₁ | C₁ | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | |
| D | D₂ | D₁ | [illegible] | [illegible] | [illegible] | | | | |
| E | E | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | | | |
| F | [illegible] | | | | | | | | |
| G | C | [illegible] | | | | | | | |
| H | [illegible] | [illegible] | | | | | | | |
| I | | | | | | | | | |
| J | | | | | | | | | |
| L | [illegible] | [illegible] | [illegible] | [illegible] | | | | | |
| M | [illegible] | [illegible] | [illegible] | | | | | | |
| N | N | [illegible] | [illegible] | | | | | | |
| | | | | | | | | | |
| P | [illegible] | [illegible] | | | | | | | |
| R | R | [illegible] | [illegible] | [illegible] | | | | | |
| S | [illegible] | | | | | | | | |
| T | T | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | | |
| V | [illegible] | | | | | | | | |
| W | [illegible] | | | | | | | | |

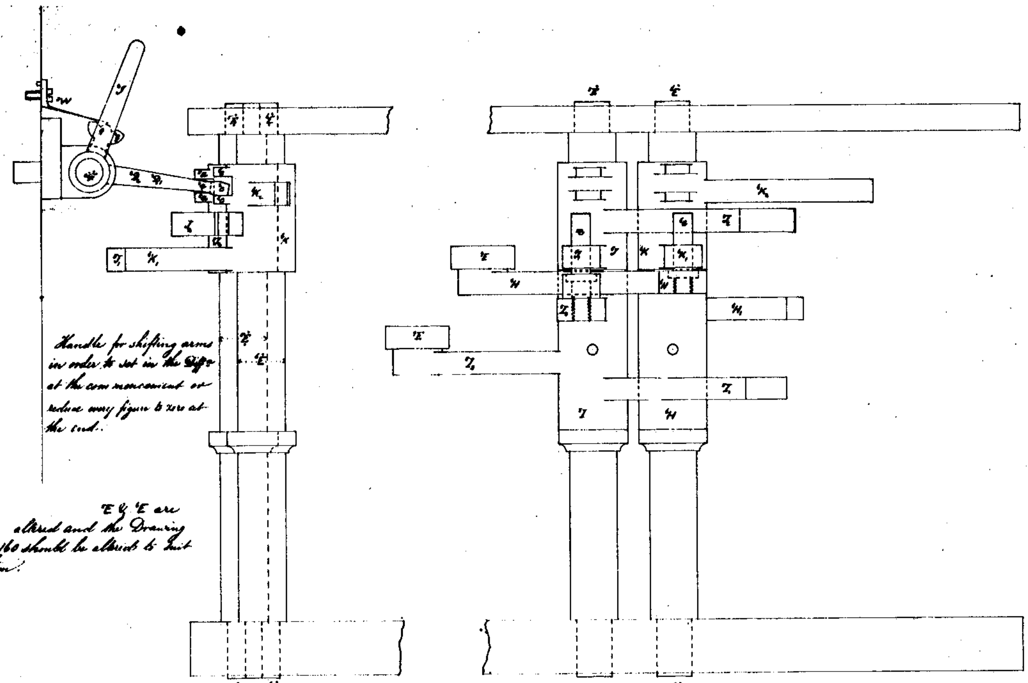



BAB [A] 166

BAB [A] 166


20
Elevation and End View of Part
of General Framing
To support one end of First
Driving Axis.
End View From Right
Elevation

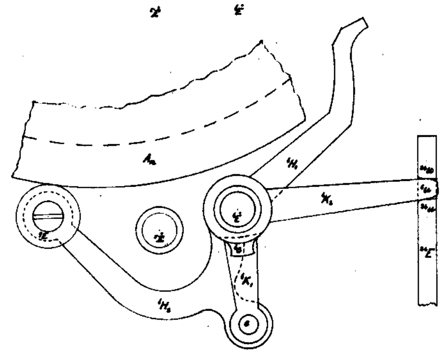
9



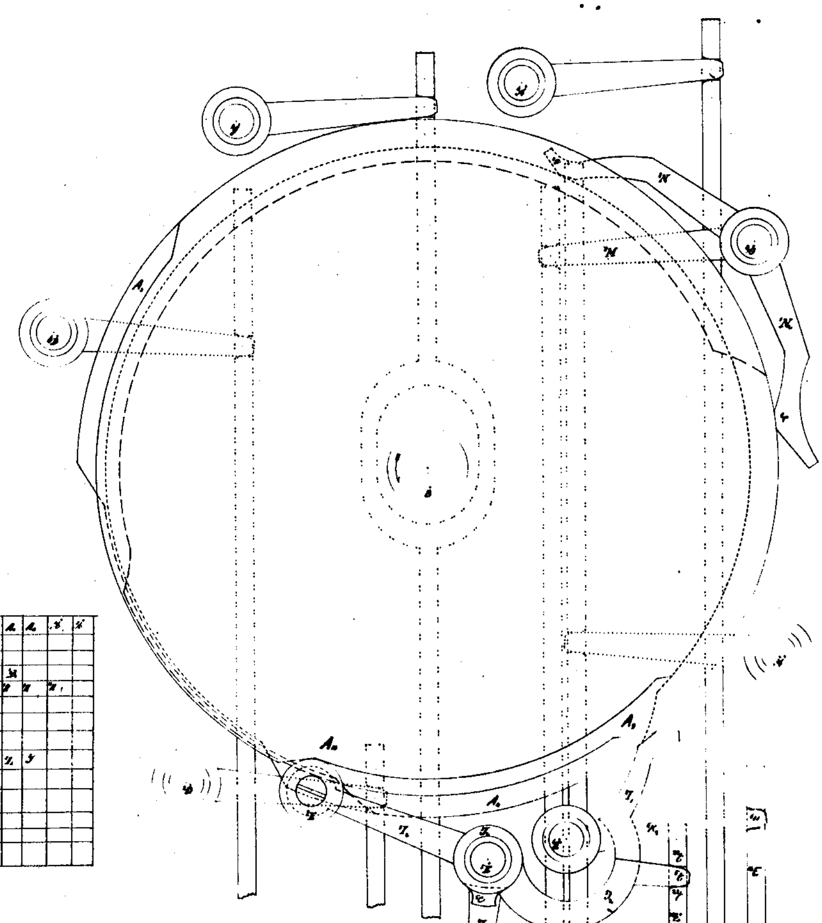
Handle for shifting arms in order to set in the Diff.s at the commencement or reduce every figure to zero at the end.

'E & 'E are altered and the Drawing H=160 should be altered to suit them.

Plan and End View of Cams &c for Vertical Motion to Calculating axes





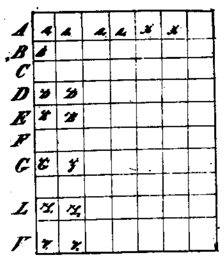


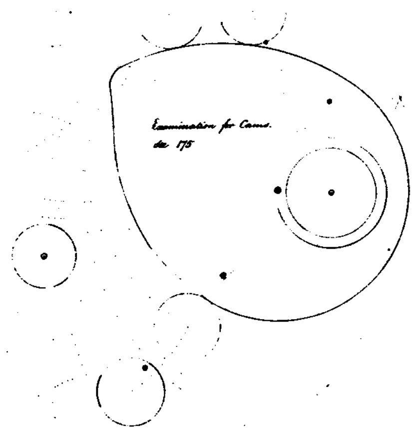
| | | | | | | | |
|---|---|---|---|---|---|---|---|
| A | A | A | A | A | A | [illegible] | [illegible] |
| B | B | | | | | | |
| C | | | | | | | |
| D | D | [illegible] | [illegible] | [illegible] | | | |
| E | E | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | |
| F | F | | | | | | |
| G | G | | | | | | |
| H | H | H | H | | | | |
| I | I | [illegible] | [illegible] | [illegible] | [illegible] | | |
| J | J | [illegible] | [illegible] | | | | |
| K | K | K | K | | | | |
| | | | | | | | |
| | | | | | | | |
| W | W | | | | | | |

BAB[A] 169

70

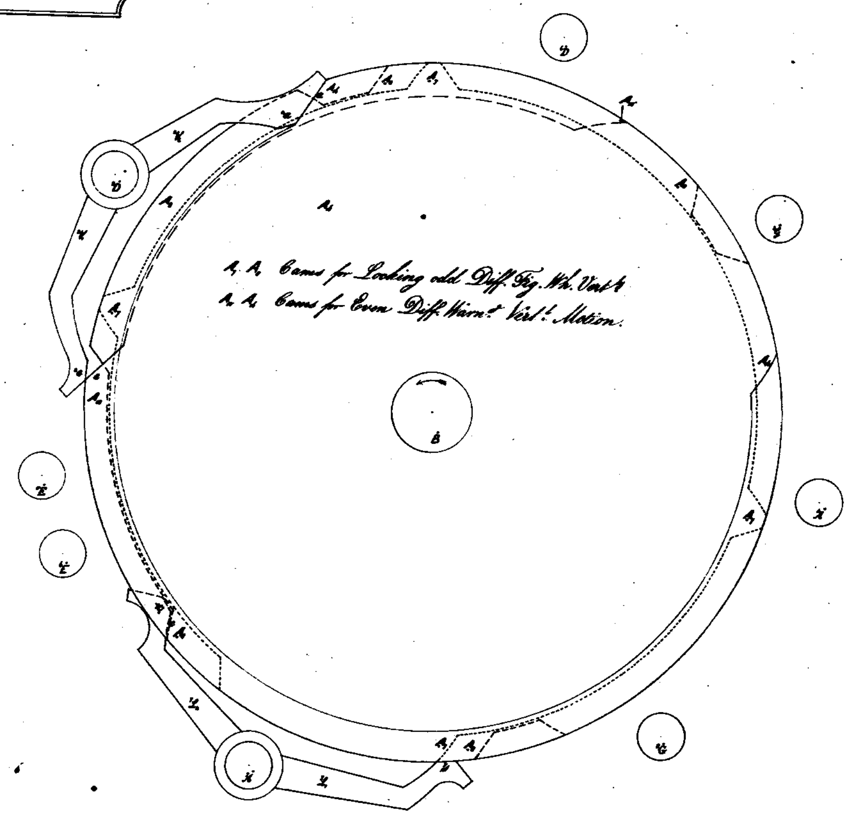
169=f



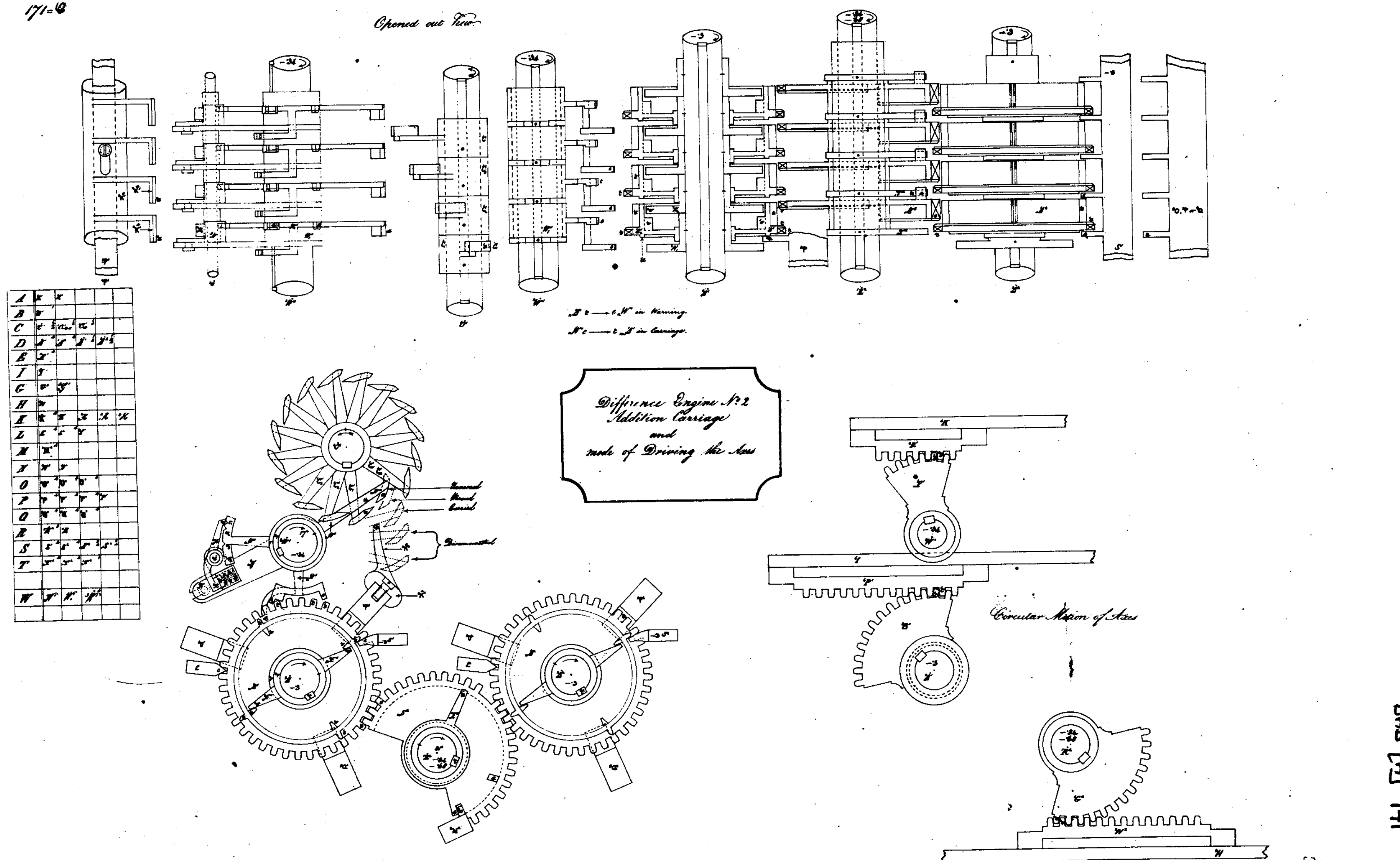



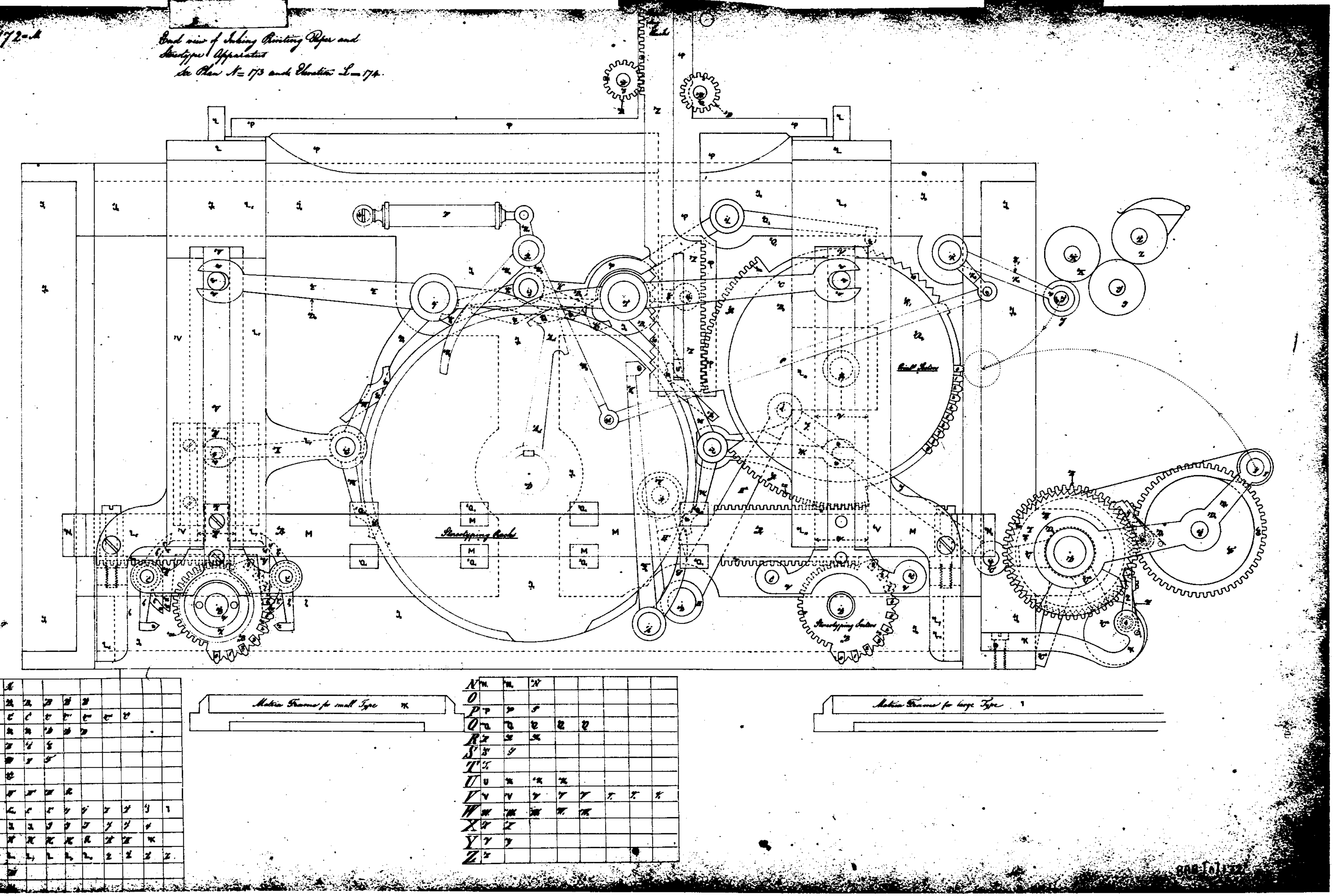

70

Plan of Cams
for Locking Odd Diffce Fig. Wheels
and for Vertical motion
of even Difference warn



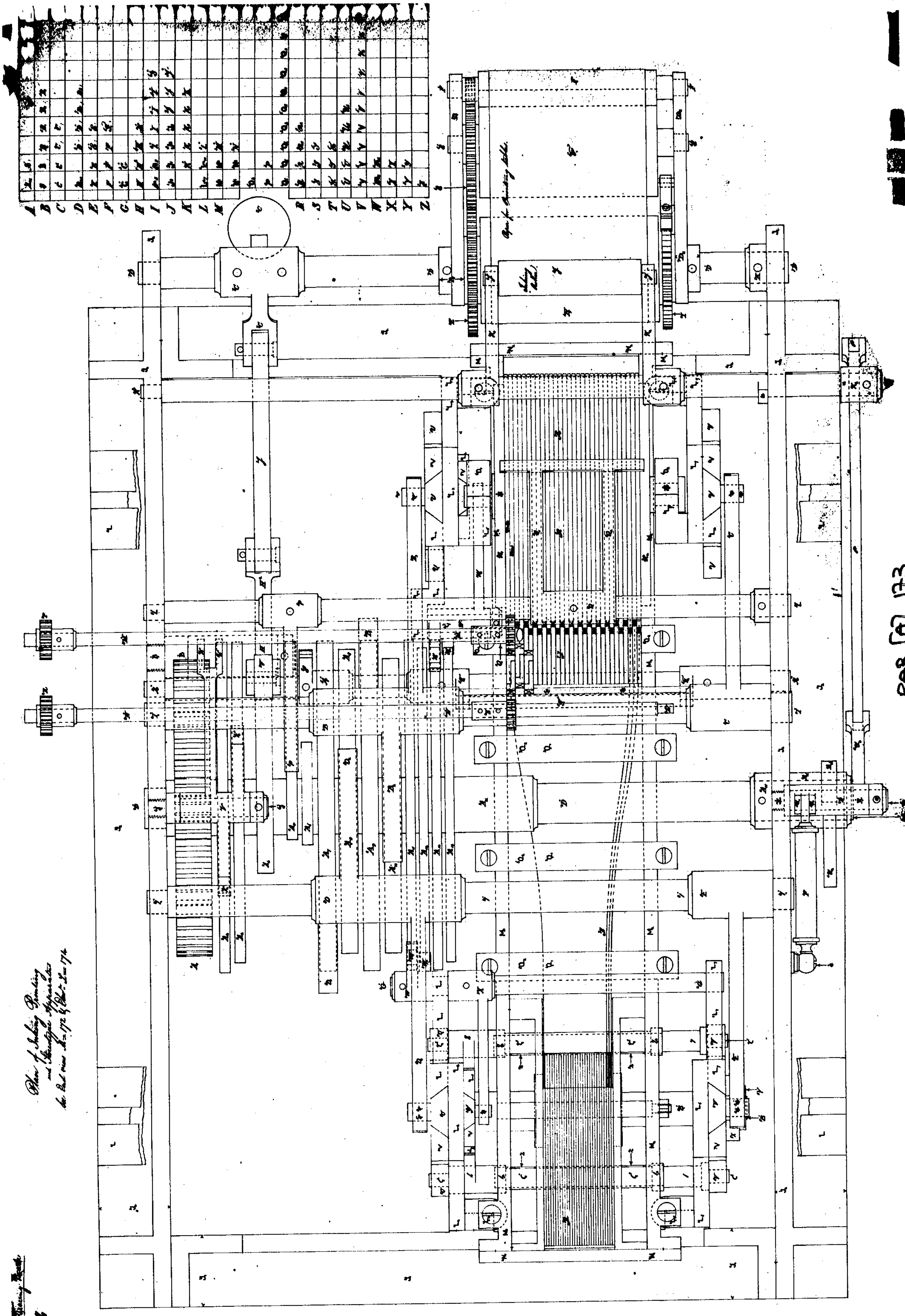

169


Opened out View
Difference Engine No 2
Addition Carriage
and
mode of Driving the Axes
Circular Motion of Axes


End view of Inking Printing Paper and Stereotype Apparatus
See Plan No 173 and Elevation L no 174.
Stereotyping Racks
Small Sectors
Stereotyping Sectors
Matrix Frame for small Type
Matrix Frame for large Type

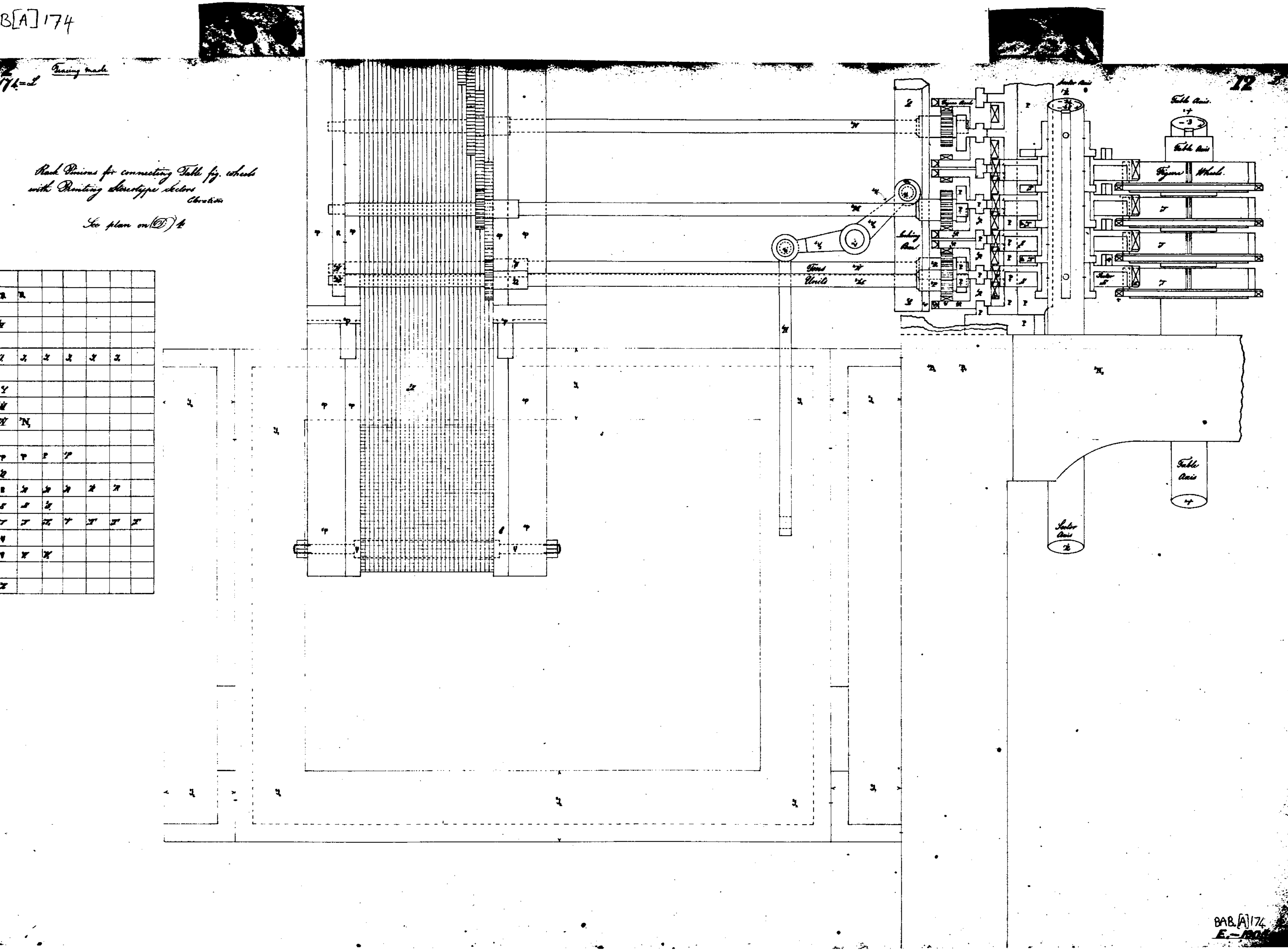

BAB [A] 172

BAB [A] 173



BAB [A] 173

BAB [A] 174



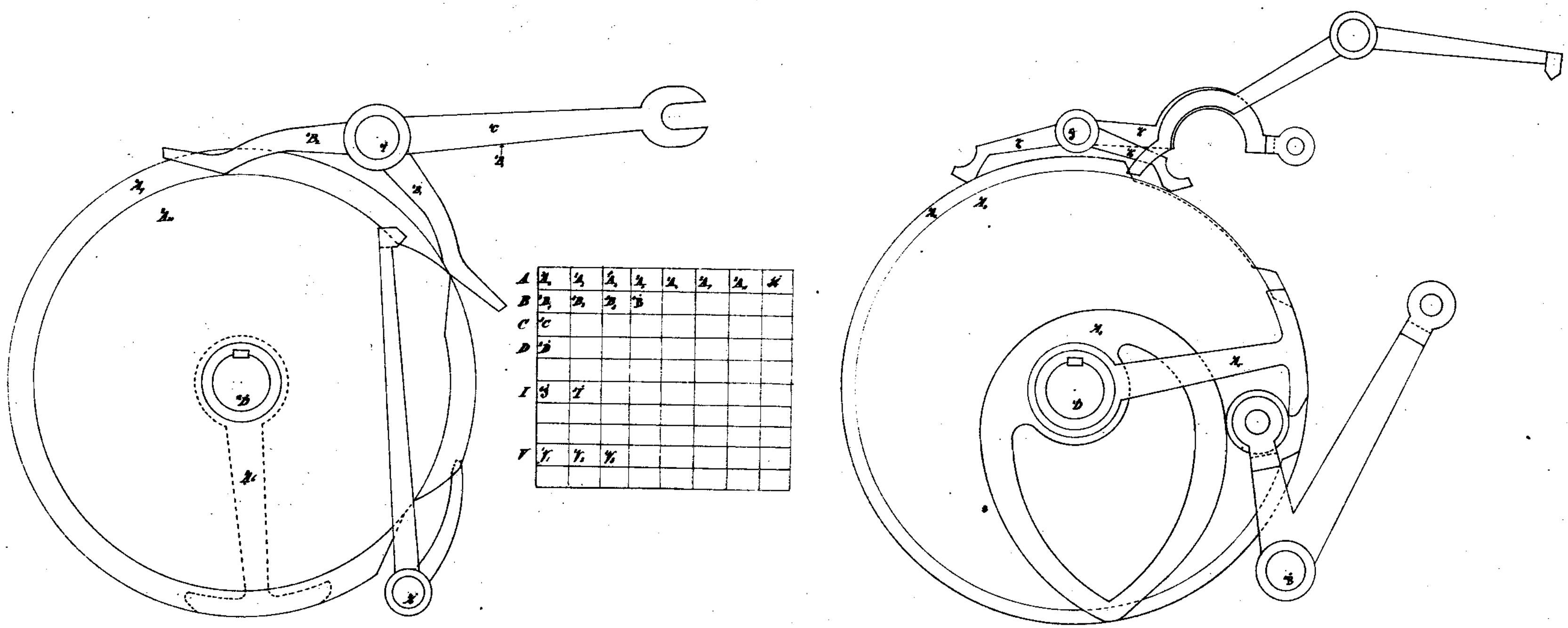

BAB [A] 174

BAB [A] 174

**17** 175=a

Plan of Cams for Punching
with small stereotype sectors, and
Cams for removing Paper Rollers

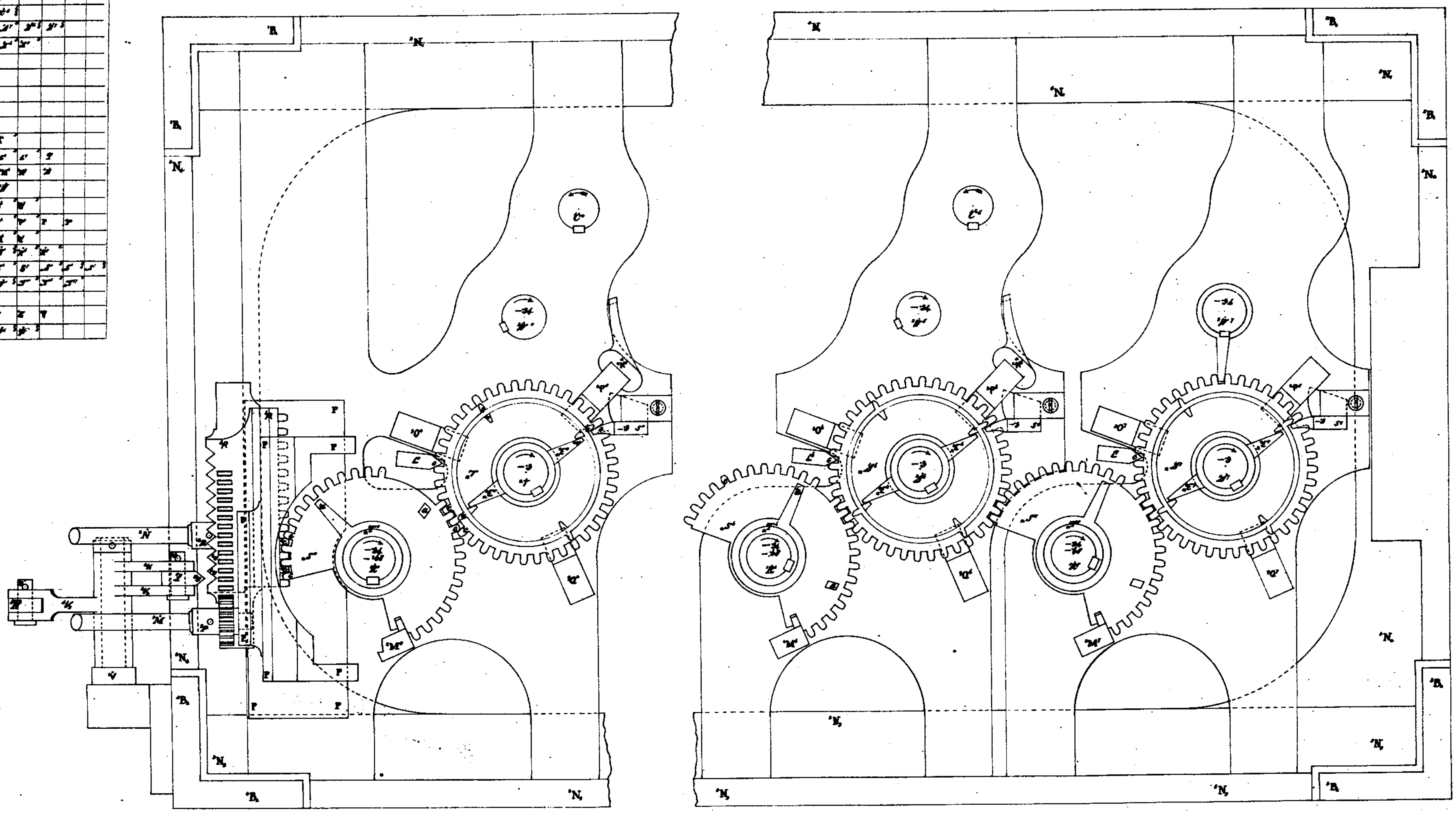
**17** a

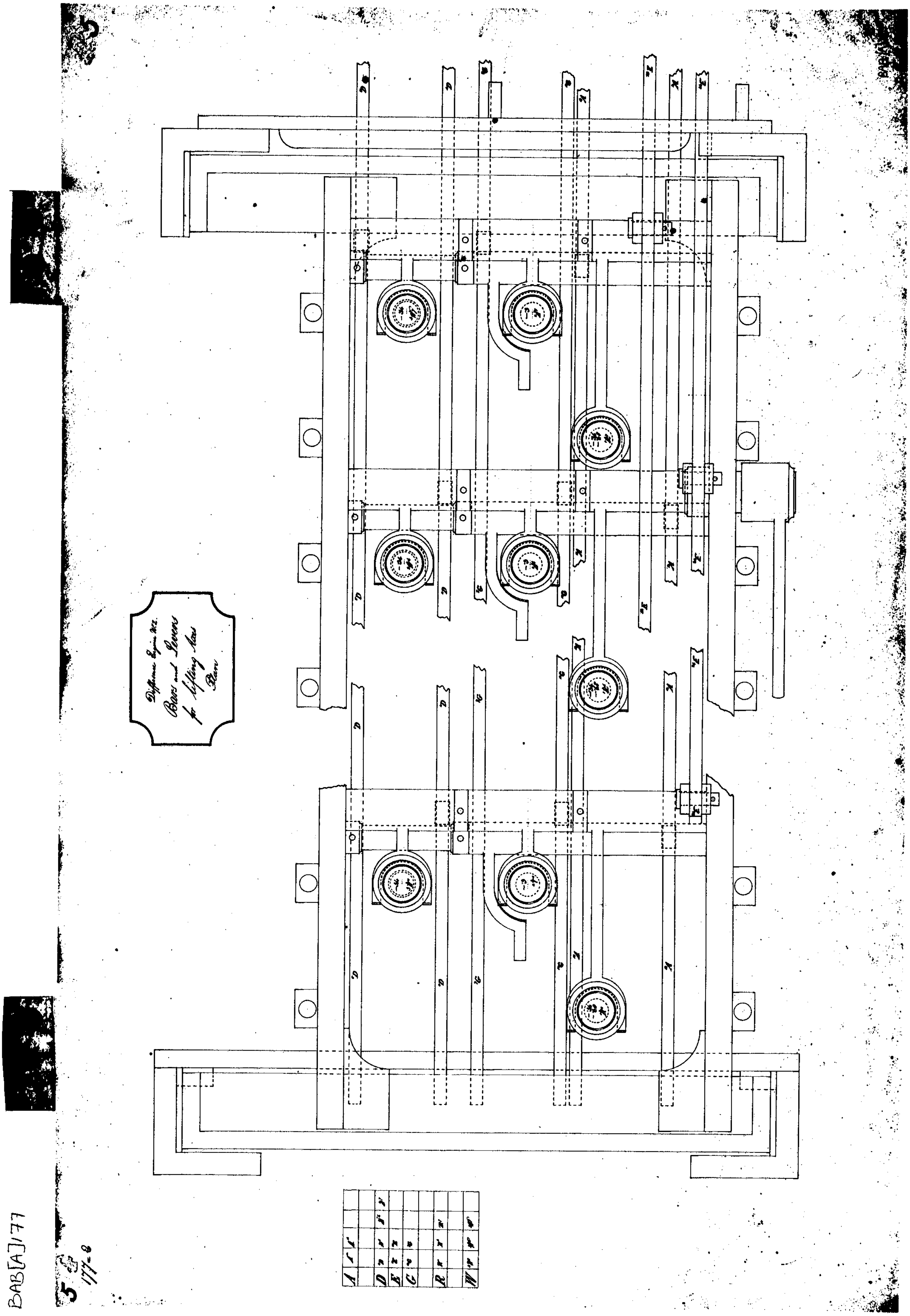


176-D Tracing made

Plan of Calculating part of Difference Engine
with the means of conveying numbers to Stereotype sectors (See Elevation in L)

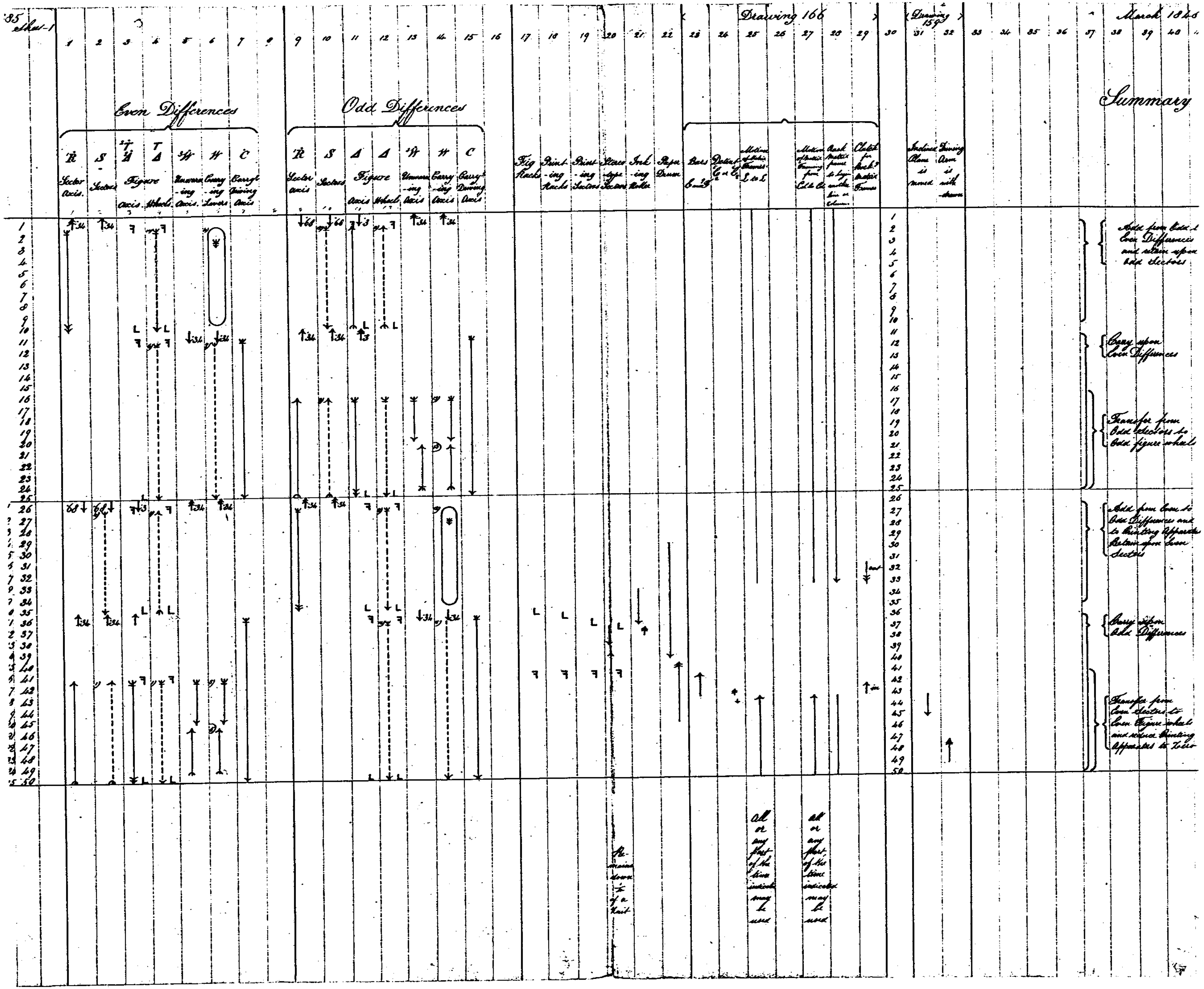
| | |
|---|---|
| A | |
| B | [illegible] |
| C | [illegible] |
| D | [illegible] |
| E | [illegible] |
| F | |
| G | |
| H | [illegible] |
| I | |
| J | |
| K | [illegible] |
| L | [illegible] |
| M | [illegible] |
| N | [illegible] |
| O | [illegible] |
| P | [illegible] |
| Q | [illegible] |
| R | [illegible] |
| S | [illegible] |
| T | [illegible] |
| U | |
| V | [illegible] |
| W | [illegible] |






BAB [A] 177

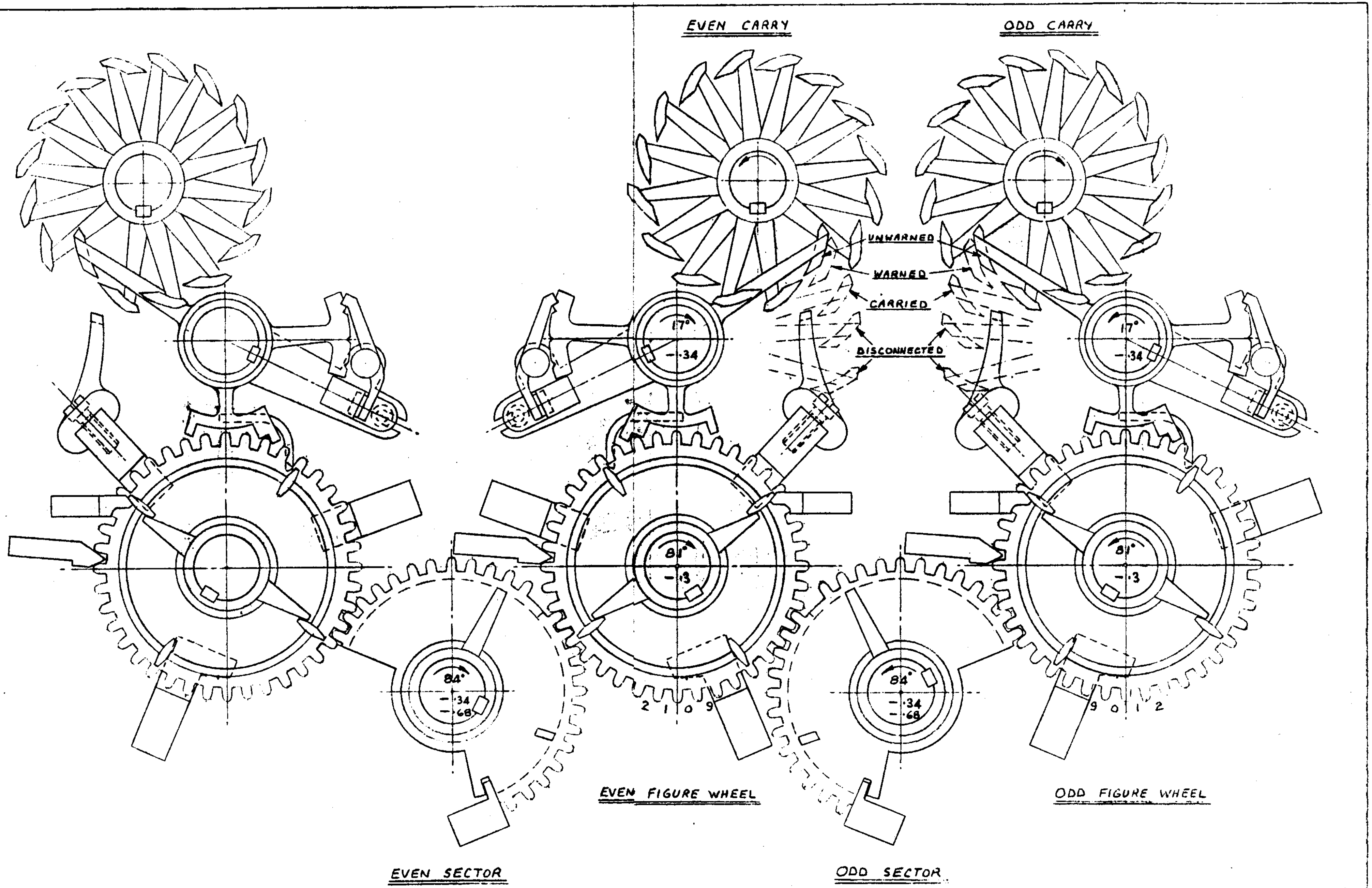
BAB[A]177






BAB [F] 385/1 [a]

BAB [F] 385/1 [a]




| MACHINE BABBAGE DIFFERENCE ENGINE No. 2 | SCIENCE MUSEUM LONDON 1991 | ADDITION CARRIAGE — SEE BAB [A]171 | SCALE 1:1 | DRG. No. 337 X 26 |
|---|---|---|---|---|