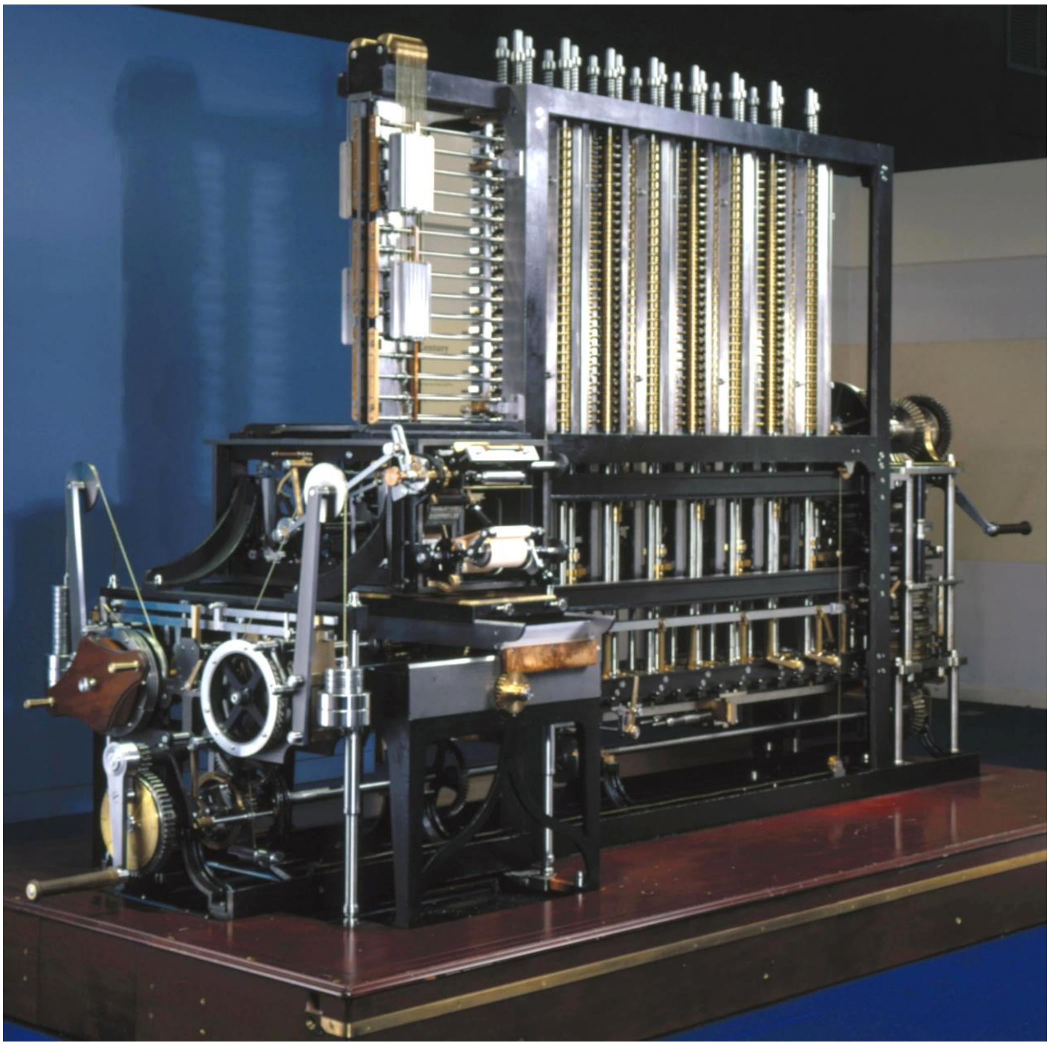
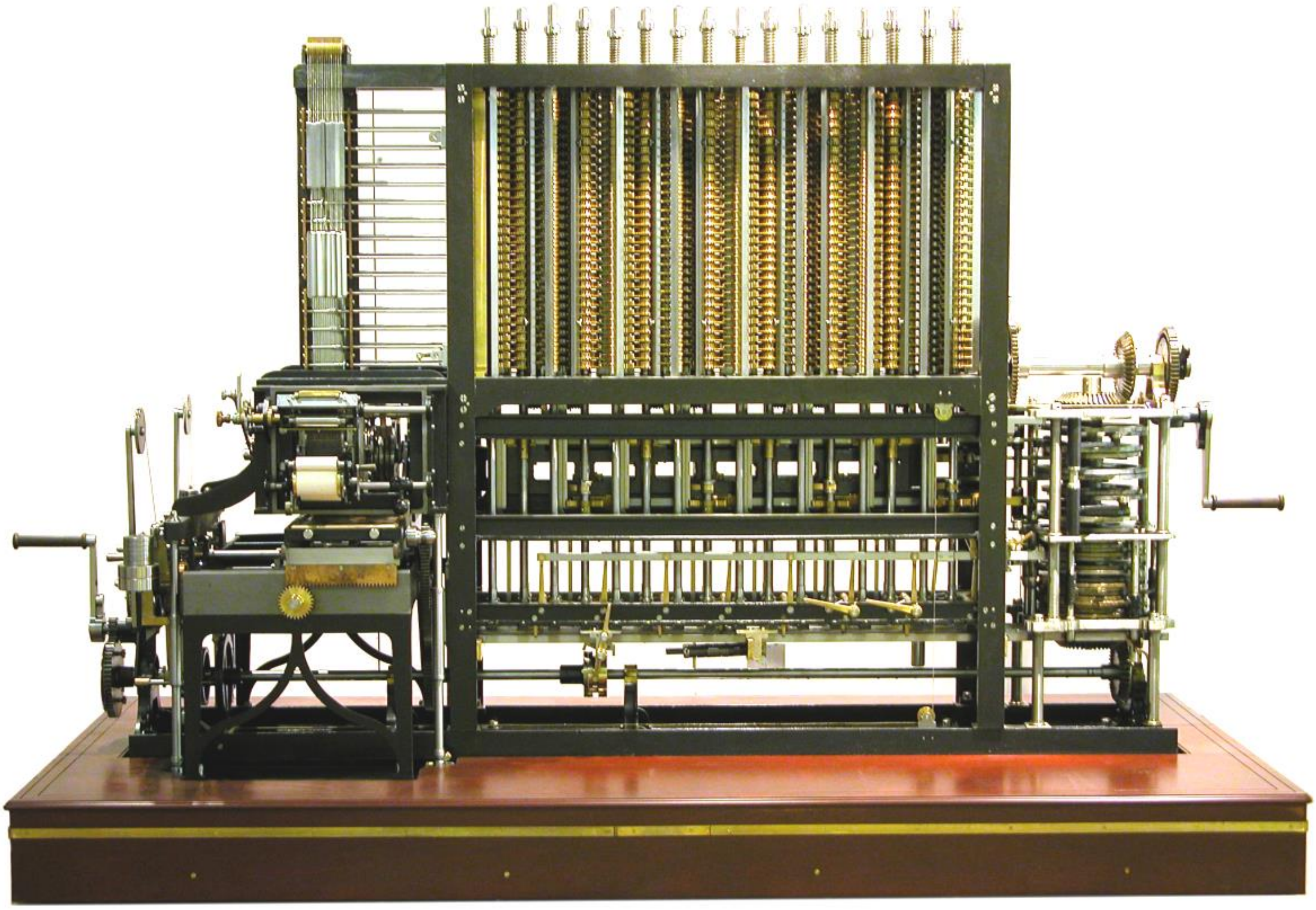
Charles Babbage's Difference Engine No. 2

Technical Description



Doron D Swade
London
March 2020



Charles Babbage's Difference Engine No. 2

Science Museum
London

Contents

Scope and Purpose

This document contains a technical description of Difference Engine No. 2 designed by Charles Babbage between 1846 and 1849, but not built till the modern era.

The account records the detailed interpretation of Babbage's designs, and documents the technical understanding that culminated in the first successful construction, completed in 2002, of an operational Babbage calculating engine built to the original drawings.

This account describes:

1. what the machine does and how it works as decoded from the 19th-century drawings
2. the rationale for design and engineering decisions that informed the modern construction
3. modifications, precautionary and remedial, made in the physical realisation of the operational engine.

This account is an historical record of the technical understanding that informed the preparation of the manufacturing drawings for the 8,000 parts of the machine.

This online version does not include images of the full set 19th-century design drawings. These are on open access and can be viewed online via the links provided. Excerpts of these drawings are included in the text so the account is for the most part free-standing. The manufacturing drawings and their companion parts lists are currently not on open access and are not included except for the modern Timing Diagram.

There is a companion document, **Charles Babbage's Difference Engine No. 2: User Manual (2020),** that provides a practical guide for the safe operation, maintenance and repair of the Engine. The User Manual is largely stripped of the interpretative explanation and design rationale included here. It is free-standing and is designed to allow responsible persons to operate the machine without necessary reference to the descriptions that follow here.

At the time of writing two working Difference Engine No. 2s have been completed. These are not production machines and there are faults that might occur that cannot be anticipated – the cumulative effects of wear, inadvertent derangement, the effects temperature variations, and breakages, for example.

Fault-finding, diagnosis and repair require an understanding of the machine and its mechanisms, and the reasons why it was implemented the way it was. This technical description is intended to aid such understanding.

List of Illustrations

4. Output Apparatus

5. Printing

6. Stereotyping

Acknowledgements

To **Sir Neil Cossons and Dame Margaret Weston**, former Directors of the Science Museum, who unhesitatingly backed the project, encouraged us, and took substantial risks in doing so. For their vision and unfailing support.

To **Nathan Myhrvold** who funded the construction of the output apparatus for the London Engine, and the construction of a complete second Engine. For his generosity, enthusiasm, vision, and support.

To the late **Allan G. Bromley** who first studied Babbage's technical designs and proposed in 1985 that Difference Engine No. 2 be built.

To the late **Reg Crick** to whom this account is limitlessly indebted. For the faultlessly systematic preparation of manufacturing drawings; for his inventiveness in devising ingenious solutions to countless design, manufacturing, and mechanical challenges; for taking the lead role in constructing the London Engine; for his limitless patience and dedication throughout the project. This volume is dedicated to him.

To the late **Barrie Holloway** who with Reg Crick specified and built the calculating section of the London Engine.

To **Richard Horton** who took on the mantle for the next generation of Babbage engineers, for assisting in the construction of the output apparatus for the London Engine; for taking the lead role in the construction of the whole of the second Engine destined for the USA; for his hall-mark perfectionism and unfailing helpfulness.

To **Chris Ward** who assisted Richard Horton in the construction of the second machine.

To **Michael Wright** for his invaluable advice and guidance on 19^{th}-century mechanical engineering, tool-making and workshop practice.

To **Tim Robinson** for his masterful care of the American Engine; for the acuity of his diagnostic, remedial and analytical understanding of the machine; for his decades-long interest and close study of Babbage's calculating engines; for heroically reading this account and providing forensically detailed feedback.

1. Introduction

The project to build a working Difference Engine No. 2 to original designs was formally proposed in 1985 by Allan Bromley, an Australian computer scientist. A team at the Science Museum, London, constructed a full working Engine that ran in its entirety for the first time in seventeen years after Bromley's initial proposal. The calculating section was completed in 1991 for the bicentenary of Babbage's birth. The output apparatus, an integral part of the machine, which prints and stereotypes results and is described in the original designs, was completed in March 2002. The project was led by Doron Swade, the author of this volume.

A second Engine was completed in 2008, also by the Science Museum, for the private collection of Nathan Myhrvold in the USA. This machine was delivered in May 2008 to the Computer History Museum, Mountain View, California, on loan from Nathan Myhrvold, for public display and demonstration there. The Engine was publicly demonstrated on a near-daily basis for some seven years – until January 2016 when it returned to Myhrvold's private collection. The two Engines were made to be identical. There are minor differences but none of these are material to the logical principles of operation or to its physical realisation. The content of this account draws on detailed analysis of the original design drawings and manuscripts dating from 1846-9, the preparation of modern manufacturing drawings that specify the 8,000 parts, and the construction and subsequent operation, over many years, of the two engines.

1.1 Brief History of Difference Engine No. 2

Genesis Episode

Charles Babbage (1791-1871) devoted the best part of his life to designing mechanical calculating engines. The genesis of his efforts is captured in the well-known vignette in which Babbage and his friend John Herschel were checking astronomical tables calculated by hand. Exasperated by the number of errors Babbage recalls that he exclaimed 'I wish to God these calculations had been executed by steam'. 'Steam' can be read as a metaphor for the infallibility of machinery, as well as a solution to the problem of supply: his calculating engine was seen as 'a machine for

Fig. 1.1: Charles Babbage (1860).

manufacturing tables' that would produce error-free tables as and when needed.[1] Babbage's mechanical epiphany occurred in 1821, probably in London. Babbage was seized by the prospects of mechanising calculation and, more generally, of mechanising mathematics.

Difference Engine No. 1

Babbage launched himself into the design of his first calculating engine. He hired a leading toolmaker, Joseph Clement, who set about making the 24,000 parts required for the whole machine. Babbage called his machine a Difference Engine, so-called because of the principle on which it was based – the method of finite differences. Practical attempts to build what later became known as Difference Engine No. 1 were abandoned in March 1833 when, after a decade of design and development, and substantial government funding, Clement downed tools and fired several of his workmen following an unresolved dispute over compensation for moving his workshop closer to Babbage's home.

During the dispute Babbage was deprived of his drawings and engine parts. Forcibly distanced from the nuts and bolts of construction he revisited some of his earliest ideas on calculating engines and conceived of a vastly more ambitious machine, his Analytical Engine – a general purpose digital programmable computing machine. He abandoned the first Difference Engine and devoted the next thirteen years to developing and refining the new designs.

Difference Engine No. 2 and the Analytical Engine

In 1846, with the major work on the Analytical Engine design done, Babbage began a design for a new difference engine, Difference Engine No. 2. The advanced functions envisaged for the Analytical Engine had made new demands on mechanical logic and design. The mechanisms for direct multiplication and division, for example, required an intricacy and complexity well beyond those for the repeated additions in the Difference Engine. He later wrote:

> … in labouring to perfect this Analytical Machine of greater power and wider range of computation, I have discovered the means of simplifying and expediting the mechanical process of the first Difference Engine.[2]

1 Babbage, B. H. (1872). Babbage's Calculating Machine or Difference Engine, Science and Art Department. Reprinted in Campbell-Kelly, M., Ed. (1989). *The Works of Charles Babbage*. London, William Pickering, Vol. 2, pp. 223-233, p. 226.

2 C. Babbage to Lord Derby, 8 June 1852. Babbage, C. (1864). *Passages from the Life of a Philosopher*. London, Longman, Green, Longman, Roberts & Green, p. 104.

The design of Difference Engine No. 2 benefited directly from the work on the Analytical Engine. The need for fast execution times led him to devise more efficient techniques for simple addition, not least to make room in the timing cycle for multiplication and division that were irreducibly more time-consuming, relying as they did on sequences of repeated operations. Meeting these demands extended his capabilities to new levels of sophistication, economy, and elegance.

Babbage is explicit on the influence of the Analytical Engine on the design of Difference Engine No 2. He wrote that he 'proposed to take advantage of all the improvements and simplifications which years of unwearied study had produced for the Analytical Engine'.[3] This was no false claim. Difference Engine No. 2 calls for some three times fewer parts than Difference Engine No. 1 for similar calculating capacity – 8,000 parts compared to 24,000.

There is a further direct connection between Difference Engine No. 2 and the Analytical Engine: the two machines share the same design for the output apparatus that prints an inked hardcopy record of results and impresses the same results into soft material in trays to create stereotype moulds from which printing plates were to be made.

The main drawings for Difference Engine No. 2 are some twenty in number and date from 1847-9. The set represents the most complete description of any of Babbage's engines. There is no evidence that any are missing or spoiled, and the drawings seem to have been spared Babbage's inveterate tinkering. It seems that for once at least Babbage was content to leave well enough alone. The design for Difference Engine No. 2 is one of elegant economy and simplicity executed by someone whose mastery of mechanical logic had already met greater challenges. The set of drawings was the primary source of technical information for the construction of the Engine.

Babbage offered the designs for the new difference engine to the government in 1852 in a letter to the Prime Minister, Lord Derby. There was evidently a legacy of discomfort from the failed first Difference Engine project some twenty years earlier: he wrote that in offering the designs of the new Engine to the government of the day he felt that he had 'discharged to the utmost limit every implied obligation I originally contracted with the country'.[4] The offer was rebuffed and Babbage retreated into bitterness and recriminations. No attempt was made to construct the machine in Babbage's lifetime. The plans were kept, and largely ignored, in a specially built wooden case, for the next hundred years.

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3 *Ibid.,* p. 97.
4 *Ibid.,* p. 107.

On his death in 1871, Babbage bequeathed the contents of his workshop and his drawings to his son, Henry Prevost, who loaned the archive to the Science Museum in 1878. The surviving physical material and drawings including those for Difference Engine No. 2, were formally donated to the Science Museum 1905.

A Modern Sequel

The Australian computer scientist, Allan Bromley, revived interest in the machine in the late 1960s when he began a detailed study of Babbage's engine designs. In 1985 he proposed to the Science Museum that Difference Engine No. 2 be built in time for the bicentenary, in 1991, of Babbage birth.

The project to build the Engine was led by the author of this account, Doron Swade, who was Curator of Computing at the time of Bromley's proposal. Metal was first cut for an experimental trial piece in December 1986, one hundred and forty years after the designs left Babbage's drawing board.

1.2 Sources

Babbage left no written explanation of his design, nor did he leave anything other than slight fragments in the way of operational instructions. His notorious thrift in the provision of textual description is true of his published writing as it is of the extensive manuscript archive that survives. His journals, called Scribbling Books or Notebooks, have intermittent relevant entries made during the design, and he left a short note on the procedure for setting up a calculation reproduced in Fig. 9.1 (p. 206). But these records are few, fragmentary and unsystematic.

Babbage's original set of drawings for Difference Engine No. 2 consists of twenty main views (average size 1000 mm by 600 mm), a small number of derivative tracings, a few superseded designs, and Notations using a language of signs and symbols of his own devising. The set of drawings is believed to be complete: there is no evidence of drawings being missing. The drawings and Notations represent the only original technical description of the engine. They are a free-standing source from which Babbage's intentions, logical and mechanical, had to be decoded and reconstructed with practically no textual explanation to assist this process. The main data source was the twenty main views: little reference was made to the derivative tracings or the superseded designs, or to the Notations other than to the timing diagram that is part of the notational description.

The drawings and Notations collectively constitute a description of the engine as Babbage intended it to be: they are sufficiently detailed to describe the shape and nominal size of individual parts and their physical relationship, and from this it was possible, with small exception, to decode intended function. However, for all the richness they contain, the drawings are not 'working drawings' in that they are not sufficiently detailed to provide a 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 the drawings provide a schematic description of logic and function, 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, for example); there are instances of omitted devices, as well as redundant assemblies. There is one screaming error that would prevent the basic addition mechanism from working. 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 a final working mechanical entity. The gap is one of engineering completeness rather than logical or operational principle: none of these practical deficiencies compromises the validity of the basic logic, design or intended function of the machine.

The manufacture of parts required full working drawings including detailed piece-part drawings specifying shape, dimensions, materials, tolerances and finish for each of the 8,000 parts. In addition, parts lists specifying quantities of individual components were needed, as well as layouts and general assembly drawings detailing the physical relationship of assembled parts.

Using Babbage's original set of drawings as the primary references 219 new A0 size drawings were produced that fully specify each of the 8,000 parts. These piece-part drawings, as well as assembly drawings and parts lists provide the necessary detail for manufacture. 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 deep knowledge of nineteenth-century practice, of Babbage's other engine designs, and of the surviving relics of his experimental assemblies. Composition analysis of contemporary 'gunmetal' was carried out to establish an appropriate grade of bronze. Dimensioning inconsistencies were resolved, incompletenesses in the design remedied, and modifications made to correct layout errors, omissions and redundant mechanisms.

Finally, the manufacturing drawings are informed by insights acquired during the course of construction, bringing the machines to a working state, and subsequent maintenance. The modern drawings include all modifications made to date, both precautionary and remedial i.e. the manufacturing drawings were updated to record modifications made during construction and testing, and following use and frequent demonstration.

The guiding principle throughout was authenticity. Care was taken to ensure that no part was specified to a higher precision than is known from measurement to have been achievable in Babbage's time. Metals were either compositionally matched with surviving examples left by Babbage in the case of bronze, and in all other instances only materials known to have been used by Babbage (cast iron and steel) of a grade available to Babbage, were specified with one non-essential exception – the grade of steel used for two impact teeth on a phasing gear (see p. 165). No attempt was made to replicate contemporary manufacturing methods, machine tools, or workshop practices. Modern manufacturing techniques were used including computer-aided manufacture but with uncompromising adherence to precision no higher than was demonstrably achievable by Babbage as evidenced in the physical relics of his own efforts.

The Mechanical Notation

Babbage used three descriptive forms to specify the Engine and its operation:

1. **Mechanical Drawings** show the shape and size of parts, and their spatial relationship to one another. (*Forms*).

2. **Symbol Strings** describe the causal chains of action of concatenated parts. (*Trains*).

3. **Timing Diagrams** show when parts move in the calculating cycle and how motions are orchestrated in timed relation to one another. (*Cycles*).

The three descriptive genres, *Forms, Trains,* and *Cycles,* collectively constitute the Mechanical Notation, a language of signs and symbols Babbage used both to describe and specify his machines, and as a design aid to optimise timing, remove redundancy and manage, symbolically, otherwise unmanageable detail at component level.[5]

[5] When he formalized the scheme in a publication in 1856, Babbage's son called timing diagrams 'Cycles'. Babbage himself was more fluid and had earlier variously referred to a timing diagram as the Notation of Periods, Notation of Units (General and Special), and Cycles of Units.

The *Forms* depict the shape and size of parts and their organisation into mechanisms. They have the form of conventional mechanical drawings and use familiar drafting conventions of plan views, front and end elevations, and sectional views in mainly third angle projections. There is nothing radical or revolutionary in these. They conform to contemporary representational conventions and capture what is essentially spatial relation.

Trains are diagrams that describe the complete causal chain from the first mover to the end result. They show the path of the transmission of motion by parts acting on other parts using the symbols and syntactical rules devised by Babbage for this purpose.

Cycles convey the orchestration of motions of individual parts into a functioning whole. For this a new set of notational conventions was introduced. Annotations at the head or tail of an arrow indicate linear or circular motion, whether rotation is positive or negative, and whether or not the motion depicted returns the part to its rest position. Other conventions indicate whether the motion is conditional or unconditional, continuous or intermittent and the time window in which the motion may or may not occur.

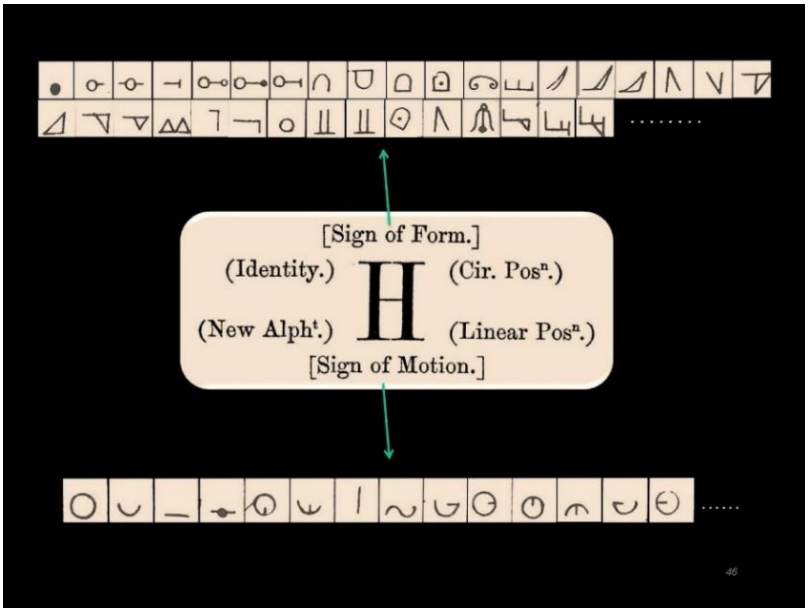

Fig. 1.2: Mechanical Notation: general template for a single part.

In the notational scheme each part is assigned a capital letter of the alphabet in one of a number of typefaces – italicised letters for moving parts and upright letters for fixed framing pieces, and a variety of typeface families are used including Etruscan, Roman, and Script. Each letter identifying a part has up to six indices – superscripts and subscripts.[6] Four of the indices are numerical – the index of identity, index of circular position, of linear position, and an index to extend the use of a typeface family in the event of running out of letters. The four numerical

[6] Swade, Doron. "'Photographing the Footsteps of Time': Space and Time in Charles Babbage's Calculating Engines." *Space, Time, and the Limits of Human Understanding*. Ed. Shyam Wuppuluri, Giancarlo Ghirardi: Springer, 2017. 417-27.

indices indicate the spatial relationship to other parts and which parts form functional groups. The two non-numerical indices are the Sign of Form, and Sign of Motion. The Sign of Form gives functional specificity to the part annotated. It indicates the species of part – rack, gear wheel, cam, pinion, arm, spring, crank, handle etc. – using symbols that are partial pictograms indicating generic function. The Sign of Motion is also part pictogram and indicates the nature of motion – linear, circular, curvilinear, reciprocating, for example.

Individual parts in the mechanical drawings are annotated with alphabetic identifiers and a selection of the non-numerical indices. The *Trains* consist of the self-same identifying letters in the mechanical drawings combined into descriptive statements using the syntactical conventions of the notational language. In the *Trains*, in addition to the numerical indices, the two non-numerical indices are extensively used to indicate class of part and the nature of motion (see, for example, BAB/A/178/1-5).

Cycles convey the orchestration of motions of individual parts into a functioning whole. For this a new set of notational conventions was introduced. Motion is indicated by arrows and variants of arrow heads, tails and shaft signify different aspects of motion: linear or circular, lifting or lowering, whether rotation is positive or negative, and whether or not the motion depicted returns the part to its rest position. Other conventions indicate whether the motion is conditional or unconditional, continuous or intermittent and the time window of possible occurrence (see, for example F/385/1 reproduced in part in Fig. 3.2, p. 21).

No significant use of the *Trains* was made in decoding the drawings for the purposes of this project. The main sources from which the operation of the machine was decoded, and from which Babbage's intentions were reverse-engineered, are the set of mechanical drawings (*Forms*) and the timing diagram (*Cycles*).

In this account extensive use is made of the same letters as in the original drawings to identify parts described in the explanatory text. The significance of the indices is, in general, not explained. On the few occasions where the Notation *is* relevant to an understanding of function, these are explained, but no comprehensive treatment of the Notation is given here. The letter or symbol for a part, with its notational indices, can be regarded simply as a non-descriptive identifier without grave penalty.

The reason that the symbolic language was not used to any great extent to inform our understanding of the design is that the Mechanical Notation had not been fully decoded at the time the drawings were first interpreted for the purposes of this project. Since then the Mechanical Notation has been the subject of a special study and in revising the account for this publication the Notation was used to revisit and verify decisions made without its earlier benefit. Where the Notation is helpful in resolving hitherto unresolved ambiguities in our original interpretation, these are noted.

1.3 Editorial Note

An express purpose of the documentation process was to capture the team's understanding of the design, to record the decisions taken in the preparation of the manufacturing drawings, and finally to document new insights and engineering modifications made during construction and commissioning. It was envisaged that the account would be of value as an historical record as well as of significant use for fault-finding, diagnostics and remedial action for those who follow.

The account that follows supersedes the print-version published in 1996: Doron Swade, 'Charles Babbage's Difference Engine No. 2: Technical Description', *Science Museum Papers in the History of Technology*, *No. 5*.

The 1996 version was written after the calculating section of the London engine was built but before construction of the output apparatus (hard-copy printer and stereotyping apparatus) was attempted. The 1996 account incorporates an analysis and description of the design of the output apparatus including precautionary modifications, but it does not include modifications made, insights gained, and lessons learned in the course of constructing the output apparatus, bringing it to working order, experimental running and maintenance. The account that follows here incorporates all technical content in the 1996 description which has been reworked and substantially expanded to include new material following the successful completion of the output apparatus in 2002, a date that marks the first successful completion in its entirety of a working Babbage calculating engine.

The 1996 description was in the nature of an interpretative companion to Babbage's original design drawings i.e. the text assumed access to the original drawings, or copies sufficiently fine to reproduce the detail of the small annotations that abound. The 1996 print document included a set of A4 drawings reduced from the large format of the originals. While the A4 drawings were adequate for much of the explanation, the fine detail, especially the minute penmanship of the Notations, did not survive reduction. Since then the original reference drawings, manufacturing drawings and parts lists have been digitised. The full set of archive drawings are available online via the links given. The manufacturing drawings are not yet in the public domain. Improvements in digital photography have also allowed more generous use of images for illustration than was possible in the 1990s.

So that the text can be at least to some extent free-standing, parts of the digitised drawings have been excerpted and incorporated in the descriptions as illustrations – this to spare the disruption and interruption of attention involved in referring to drawings whether digital or print. Where reference is made to parts or mechanisms of which there is no substantial discussion, or the mention is in passing, the relevant drawings-excerpts have not been incorporated in the text and the full drawing will need to be referred to.

References are made to manufacturing drawings particularly in the discussion of modifications, and also to describe aspects of the general assembly. These manufacturing drawings are not yet on public access. Since the principal purpose of this account is to chronicle our interpretation of the original twenty 19th-century drawings, no comprehensive description of the modern drawings is undertaken here but reference to them is made by way of supplementary description.

The text is also illustrated with digital photographs of parts of the built machine. I confess to a methodological discomfort here. A purist approach would be to present, as an act of scholarly transparency, the analysis and interpretation of the drawings as the rationale and justification for the physical implementation. Using images of the outcome to illustrate the analysis would therefore seem to beg the question (in the original, logically circular, sense of this phrase) i.e. the circularity entailed by using the outcome as part of the justification for the process by which the outcome was achieved.

When the documentation project was conceived the Technical Description was envisaged as a text-based account largely in the nature of an archival companion to the physical drawings. But with advances in digital photography the temptations to use images of built parts to illustrate the analysis proved too strong to resist. These images are used purely to aid understanding and it is hoped that their use will not in any way lessen critical examination of the interpretation of the original drawings presented here.

Countless times during this project we lamented Babbage's reticence in providing any written explanation of his drawings, intentions, and designs. For those studying the Engine, now or in the future, it is hoped that the account that follows will, at least to some extent, spare us similar censure.

Doron Swade
London
February 2020

2. Overview

Difference Engine No. 2 automatically calculates and tabulates a class of mathematical functions (polynomials), prints results in inked hardcopy, and impresses results in trays of soft material to create moulds (stereotypes) for the production of printing plates. The Elevation (Fig. 2.1) shows the overall form of the Engine.

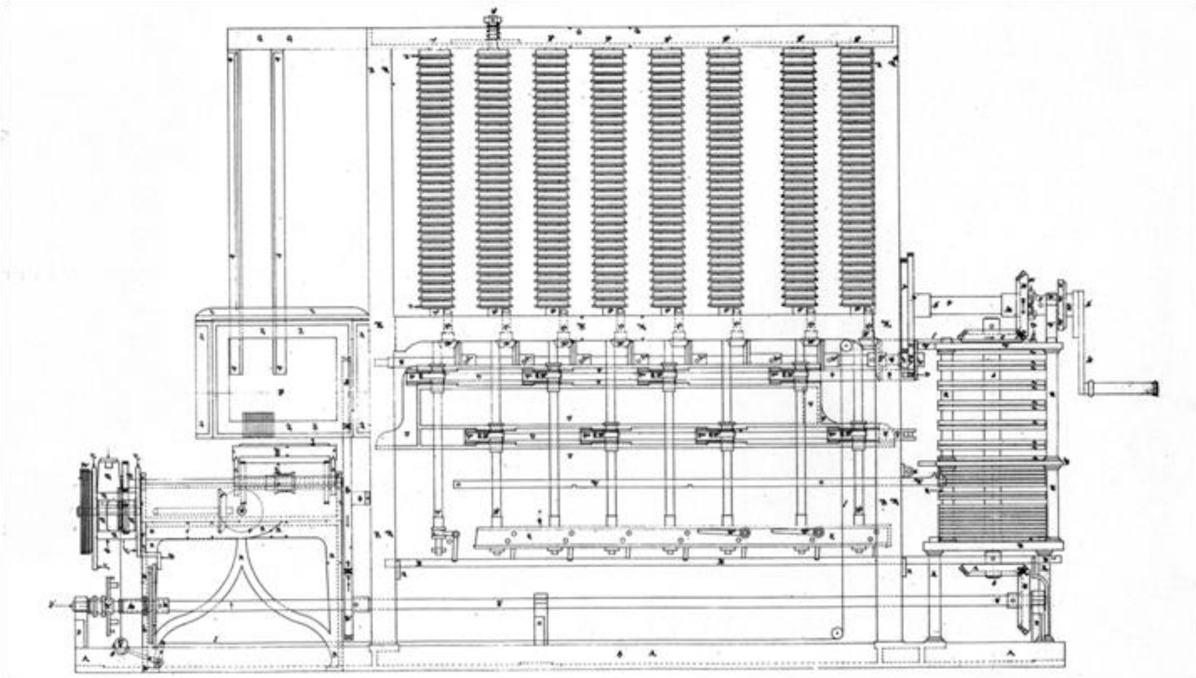

Fig. 2.1: Difference Engine No. 2, Main Elevation, c. 1848

The whole machine measures eleven feet long and seven feet high with the depth varying between eighteen inches and four feet (Fig. 2.2). The built Engine weighs 5 tonnes and consists of a total of 8,000 parts made of bronze, cast iron, or steel.

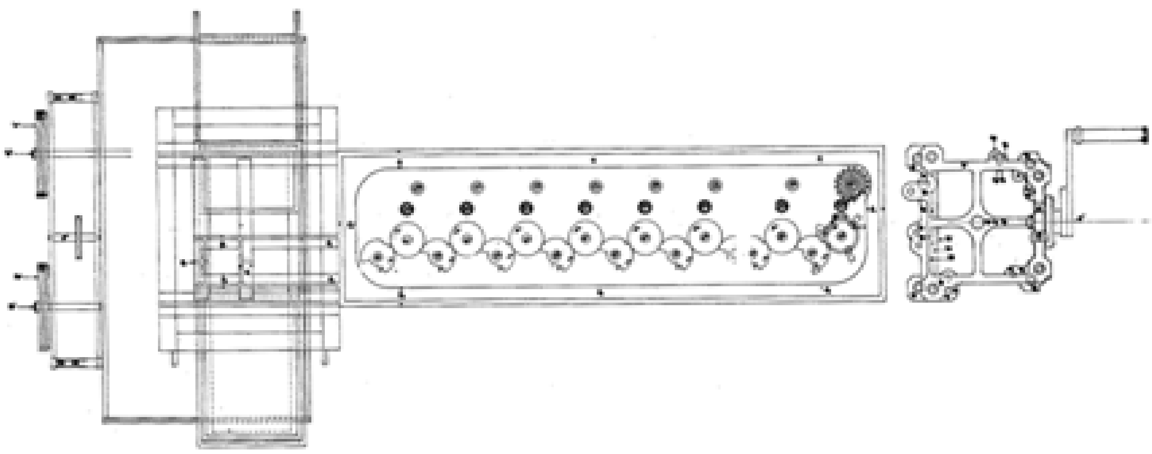

Fig. 2.2: Difference Engine No. 2, Plan, c. 1848 (A/164)

The Engine calculates and tabulates any 7th-order polynomial to thirty one decimal places, and prints inked hard copy and stereotypes to thirty decimal places. It is operated by turning the crank by hand (Figs. 2.1, 2.2) and each full cycle of the Engine produces the next result in the table.

The Engine has three main sections bolted together to form a monolithic whole:

1. Control Unit
2. Calculating Section
3. Output Apparatus

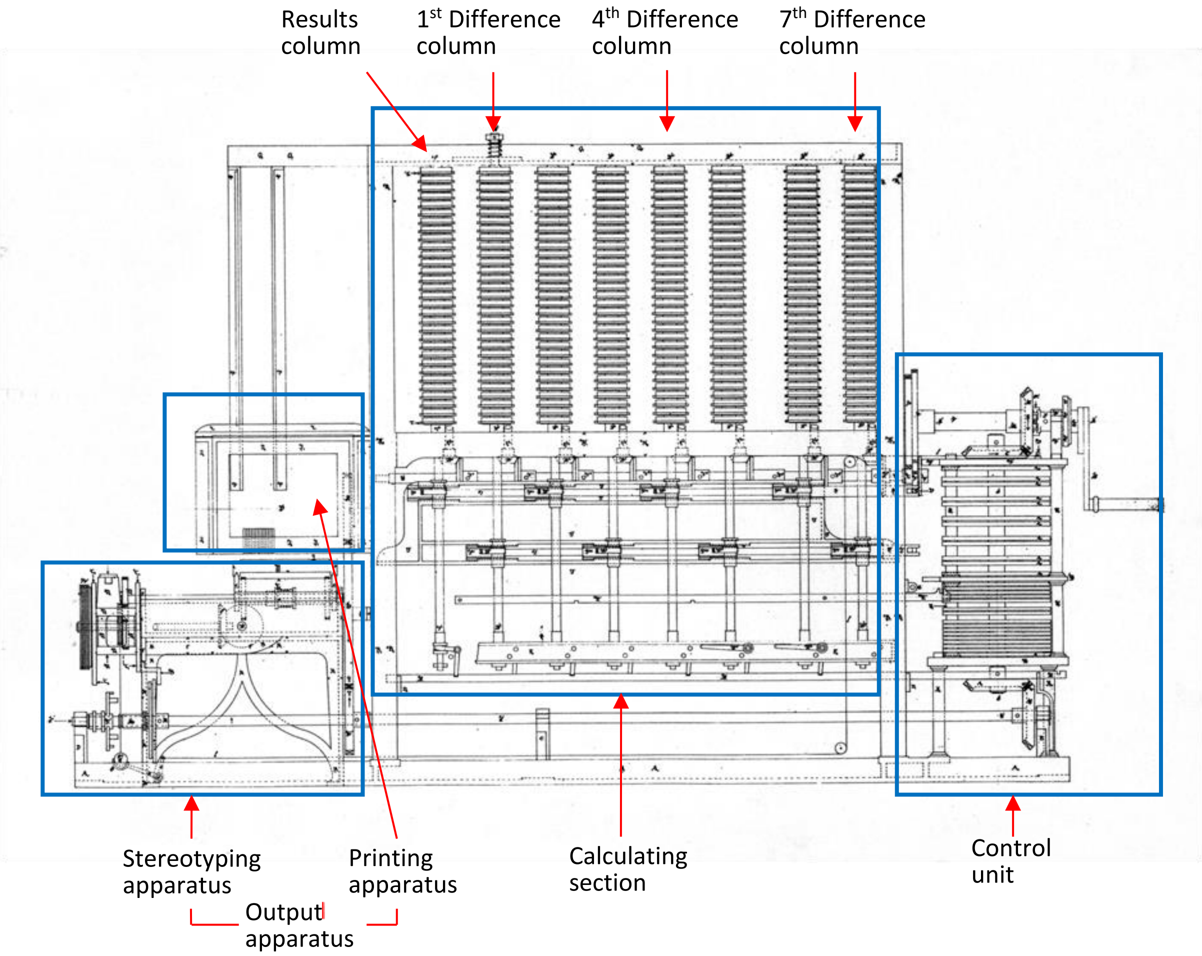

Fig. 2.3: Difference Engine No. 2, Main Elevation showing three main subsections (A/163).

2.1 Control Unit

The Control Unit consists of the main crank and the cam stack. The Engine is operated by turning the crank by hand. The crank drives a set of twenty-eight cams (fourteen pairs) arranged in a vertical stack shown immediately alongside the crank. The rotating cams drive and synchronise the lifting, turning and sliding motions required by the calculating mechanism.

The crank handle also drives the printer and stereotyping apparatus through a long shaft running the length of the underside of the machine and driven by a large bevel gear on the underside of the cam stack (Figs. 2.1, 2.3).

The output apparatus has its own local control unit in which a vertical set of cams controls the internal operations and timing of the printing and stereotyping mechanisms.

2.2 Calculating Section

The Calculating Section consists of two parts: the upper section features eight columns of figure wheels; the lower section consists of the drive links, levers, racks and pinions that produce the lifting and turning motions for the vertical axes. The mechanisms are driven by the rotation of the cams in the cam stack. The calculating section and control unit together consist of 4,000 parts.

Each of eight 31-digit numbers (seven differences and the tabular value), is represented by a column of thirty-one figure wheels, one for each digit of a multi-digit number. The number value of each digit is represented by the angular rotation of a figure wheel. Each wheel has forty teeth with the pitch of one tooth corresponding to a single decimal number, 0 to 9. So each figure wheel has four identical runs of ten decimal numbers engraved on the barrel and which can be read off directly. (Fig. 2.4).

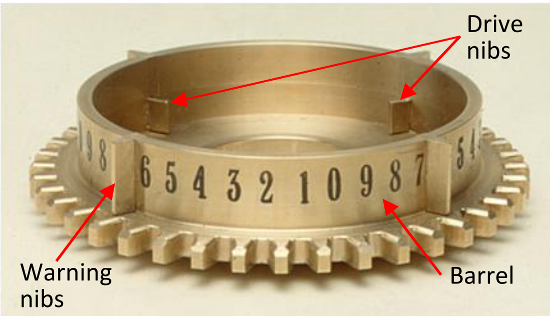

Fig. 2.4: Figure wheel. One of 248.

In each column the least significant digit is at the bottom, the most significant digit is at the top i.e. units are represented by the lowermost figure wheel, tens the next wheel up and so on. The right-most column holds the value of the 7^{th} difference; the 6^{th} difference is on the column immediately adjacent to the left and so on. The tabular result appears on the left-most column, the results column (Fig. 2.3)**Error! Reference source not found.**.

Digit values of results and differences can be read directly from the figure wheels. Immediately above each wheel is an engraved cursor facing front, that acts as a pointer. The cursors are used to read digit values when setting up initial values at the start of a calculation, verifying correct operation and fault-finding. Throughout this account the front of the Engine consistently refers to the view of the machine that allows digit values, indicated by the cursor, to be read from the figure wheels (See **User Manual (2013), Calculating Section,** Fig. 2.6, p. 10.)

The 7th, 5th, 3rd, and 1st difference columns are the odd difference columns. The 6th, 4th, and 2nd difference columns are the even difference columns.

In the case of a full 7th-order polynomial, for each calculating cycle, the Engine is required to add the value on the 7th difference column to the value on the 6th difference column, the 6th to the 5th and so on with information moving from right to left as seen from the front of the Engine, as shown in Figs. 2.1, 2.3. The result of these additions, the tabular value, appears on the last column on the left, the results column. However, the Engine does not carry out the addition of differences in this strictly serial way. Instead values of the four odd differences columns are added to the even difference columns in the first half-cycle and the even differences are added the odd differences in the second half cycle in a process akin to 'pipelining' in present day systems. The operation and rationale for this sophistication is described in **3.5 'Pipelining'**, p. 45.

The initial values for a calculation are entered by hand using a fixed sequence of steps, the setting up procedure. Initial values are entered on the figure wheels from a pre-prepared table calculated specifically for the function being tabulated. After setting initial values, each subsequent cycle of the Engine produces each next value in the table of the function being tabulated.

The tabular value on the results column is automatically transferred to the output apparatus on the left for printing and stereotyping (Fig. 2.3). Only thirty of the thirty-one digits of the result are transferred to the printer so digit thirty (the highest value digit) will not be printed or stereotyped.

The machine can calculate and tabulate any polynomial up to the 7th order using repeated addition according to the method of finite differences. For polynomials of order less than seven, the higher order difference columns are set to zero and play no part.

Integrity of the Calculation

Babbage's initial motivation for mechanising tabulation was to eliminate the risk of error in the manual production of printed mathematical tables. Machines are not inherently error-free in virtue of being mechanical i.e. they are not immune to malfunction, derangement, wear or breakage, and the designs feature a variety of security devices – mechanisms to prevent derangement, for error detection and correction – to ensure the integrity of the calculation. Babbage boasts that with these security devices the Engine will break, jam but never deceive.[1]

The Engine is a decimal digital machine: it uses the familiar decimal number system, and it is digital in the sense that only discrete integers are legitimate representations of number values. The measures for error prevention, detection and self-correction rely on the feature that only discrete intervals of a part's motion are valid representations of number values.

The basic calculating unit, a figure wheel, is not inherently digital: it has a fixed number of teeth with each tooth-interval representing a single integer, but the angular position of a wheel is not discretised, that is, the wheels can take up any rotational position between integral whole numbers, and are capable of continuous motion with all transitional states between integers physically viable. This is unlike an electronic flip-flop that flips from one state to another when a threshold is exceeded and in which the transitional states are not stable. A toothed figure wheel at rest in an intermediate position between two number values is indeterminate and the occurrence of this state during a calculation signals that the integrity of the calculation has been compromised.

Digital operation is achieved by locking and control mechanisms rather than any inherent discreteness in the motions of the basic mechanical elements or mechanisms. Locking and security devices to ensure digital control are essential features of the design.

2.3 Output Apparatus

The output apparatus (Fig. 2.5) automatically typesets results, prints an inked copy of each result on a paper roll, and produces stereotype moulds by impressing the results into soft material (wet Plaster of Paris, for example) held in trays (Figs. 2.6, 2.7).

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[1] Babbage, C., *On the mathematical powers of the calculating engine*, 1837. Campbell-Kelly, M., ed. *The Works of Charles Babbage*. 1989, William Pickering: London, Vol. 3, p. 39.

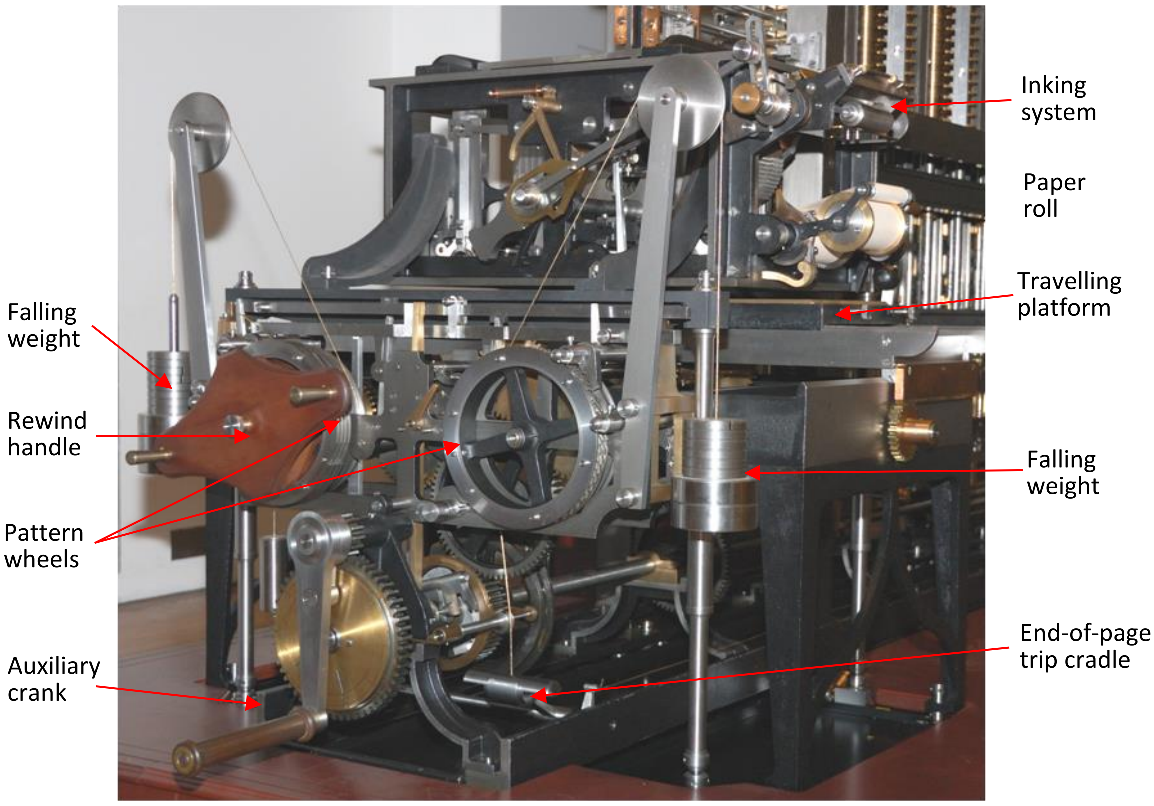


Fig. 2.5: Output Apparatus for printing and stereotyping.

The apparatus consists of two sections (Fig. 2.3): the printing apparatus, which includes the inking apparatus, paper roll, and printing wheels; and the stereotyping apparatus immediately below it. The whole apparatus consists of 4,000 parts, is bolted to the main frame of the calculating section (Fig. 2.5) and is an integral part of the original design.

The trays are fixed to travelling platforms that automatically position the them under the punch wheels to receive each new result. One of the trays is larger than the other and there are two sets of punch wheels with different font sizes, one set for each tray. Punch

Fig. 2.6: Inked hardcopy (experimental).

Fig. 2.7: Stereotype moulds in Plaster of Paris.

wheels are lowered once each cycle to create an impression of results, and the resulting tablet is used as a mould to make a printing plate (Fig. 2.7).

The format of the stereotyped results can be programmed: combinations of line height, number of lines, margin widths, and number of columns can be preset using pattern wheels (Fig. 2.5). Blank lines can be left between groups of lines for ease of reading.

The stereotyping apparatus can impress results across the page (column-to-column) or down the page (line-to-line). In the case of column-to-column format, line wrap at the end of the line is automatic. Similarly, in the case of line-to-line format, fly-back to the top of page at the end of a column of results is automatic.

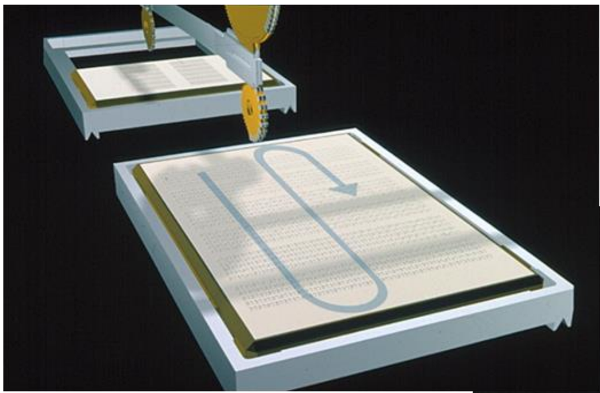


Fig. 2.8: Line-to--line format with automatic fly-back.

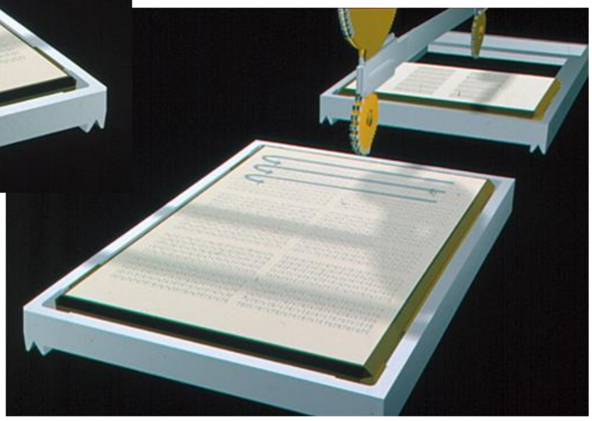


Fig. 2.9: Column-to-column format with automatic line wrap.

The output apparatus is directly coupled to the calculating section and each new thirty-digit tabular value is transferred automatically from the results column, via a system of racks, pinions and spindles, to the output apparatus for printing and stereotyping. There is no printing overhead i.e. each result is printed and stereotyped during the calculating cycle that generates it with no buffering or any additional time taken.

The output apparatus prints and stereotypes thirty of the thirty-one digits of the result. The highest value digit, the topmost figure wheel in the result column, is not transferred and therefore not printed or stereotyped.

The whole output mechanism is driven from a main drive shaft running along the underside of the Engine from the control unit (Fig. 2.1). There is a separate set of cams local to the output apparatus that drives the internal motions of parts and orchestrates the timing of operations for printing and stereotyping.

A feedback mechanism from the output apparatus to the calculating section automatically halts the Engine at the end of a stereotyped page to prevent overrun. End-of-page halting ensures that the first new result on the fresh tray is the next result in the sequence and that no results are lost in the changeover.

The travelling platforms bearing the two stereotyping trays are driven by two sets of falling weights. One set (on the right in Fig. 2.5) drives the line-to-line movement. The set on the left drives the column-to-column movement.

When stereotyping down the page (line-to-line) the line-to-line falling weight is automatically rewound and a new column can be started without interruption. At the end of a full page the engine halts and the column-to-column falling weight is rewound by hand using the rewind handle (Fig 2.5). When stereotyping across the page the column-to-column falling weight is rewound automatically and at end-of-page the line-to-line falling weight is rewound by hand.

3. Calculation

The mathematical principle on which the Engine is based is the method of finite differences. This method requires the successive addition of the highest order difference to the next lower order difference, which is itself added to the next lower order difference and so on, until the first difference is added to the tabular column to produce the next value of the expression being tabulated. The highest order function that can be tabulated is a 7th-order polynomial.

Mapping this onto the Engine (Figs. 2.3, 3.1) the result (tabular value) appears on the left-most column, the one closest to the output apparatus. The 7th difference is held on the right-most column and the first difference on the column immediately to the right of the results column. The digit wheel at the bottom of a column represents the least significant digit (units), tens are the next decade up, hundreds the next, and so on.

There are four odd-difference columns (7th, 5th, 3rd, and 1st) and three even difference columns (6th, 4th, 2nd differences) plus the results column which has the configuration of an even difference column. Whether a column holds an odd or even difference is significant because of the way the timing cycle is phased: the Engine uses a more sophisticated sequence than strict serial addition of one column to the next (see **3.5 'Pipelining'**, p. 45).

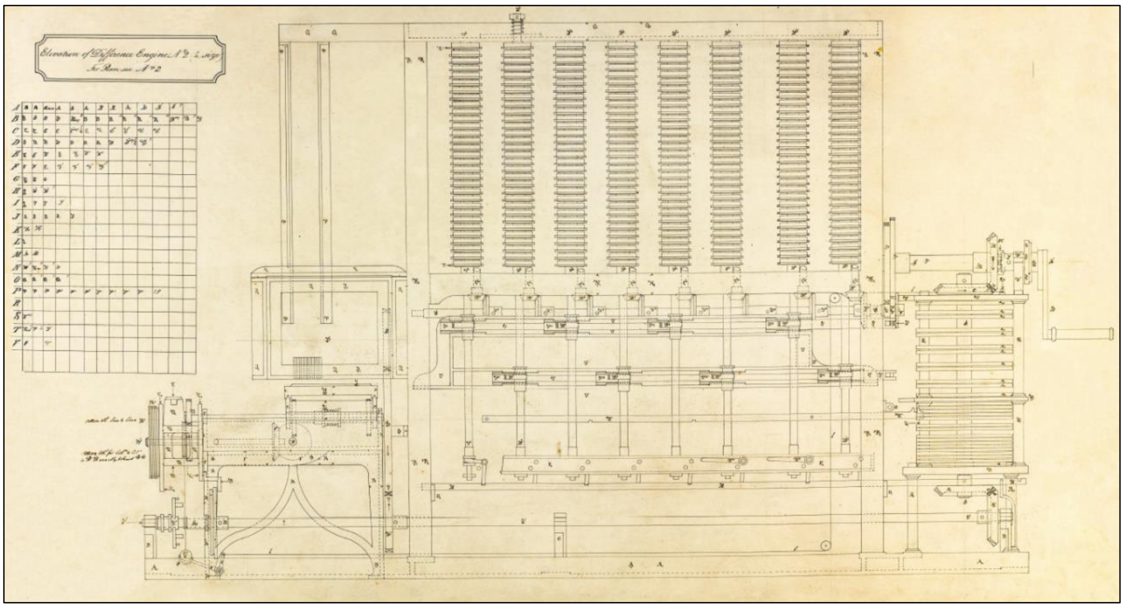


Fig. 3.1: Difference Engine No. 2, Main Elevation showing general layout (A/163).

For calculations requiring seven orders of difference all eight columns are active and the 7th difference column holds the constant difference. In situations in which there are fewer than seven differences the active columns are those closest to the tabular value column, and columns closest to the drive handle are inactive, that is to say, all thirty-one figure wheels in each of the unused columns column are set to zero. The internal axes of the inactive columns are still driven but with values set to zero they have no effect on the calculation. For a pure 7th-order polynomial the 7th difference value is constant. In cases where fewer than seven differences are used, the highest order difference column (the right-most active column) holds the constant difference.

The addition of a numbers as implemented in the Engine consists of four distinct operations:

1. **Giving-off** - the process by which the number on one figure wheel is added to the adjacent (lower) difference figure wheel in the column alongside. The wheel giving-off reduces to zero and the number on the adjacent wheel is increased by the number given off.

2. **Warning** - the action of setting a warning latch that a carriage is still outstanding when a number value on the receiving wheel exceeds 9 during giving-off. The latch warns that after giving-off, the next higher decade needs to be incremented by one unit to complete the addition.

3. **Carriage** - the carriage of tens in which a number value on a figure wheel is incremented by one, if the figure wheel immediately below it is warned.

4. **Restoration** - restoring the number given off for use in the next calculating cycle.

Timing and Phasing

The timing sequence for a full calculating cycle (two half-cycles) is described in an original Timing Diagram F/385/1 (Fig. 3.2).

Babbage divided the calculating cycle into fifty time units so each of Babbage's units corresponds to 7.2° in a 360° cycle. The timing units are shown numbered in the left-hand column in the Timing Diagram (Fig. 3.2). The columns across the top are captioned with the names of the moving parts active in the calculation and their notational identifiers: Sector Axis, Sectors, Figure Wheels and Figure Wheel Axis, Unwarning Axis (called Warning Axis in this account), Carrying Levers, and Carry Driving Axis (called Carry Lever Axis, or simply Carry Axis, in this account).

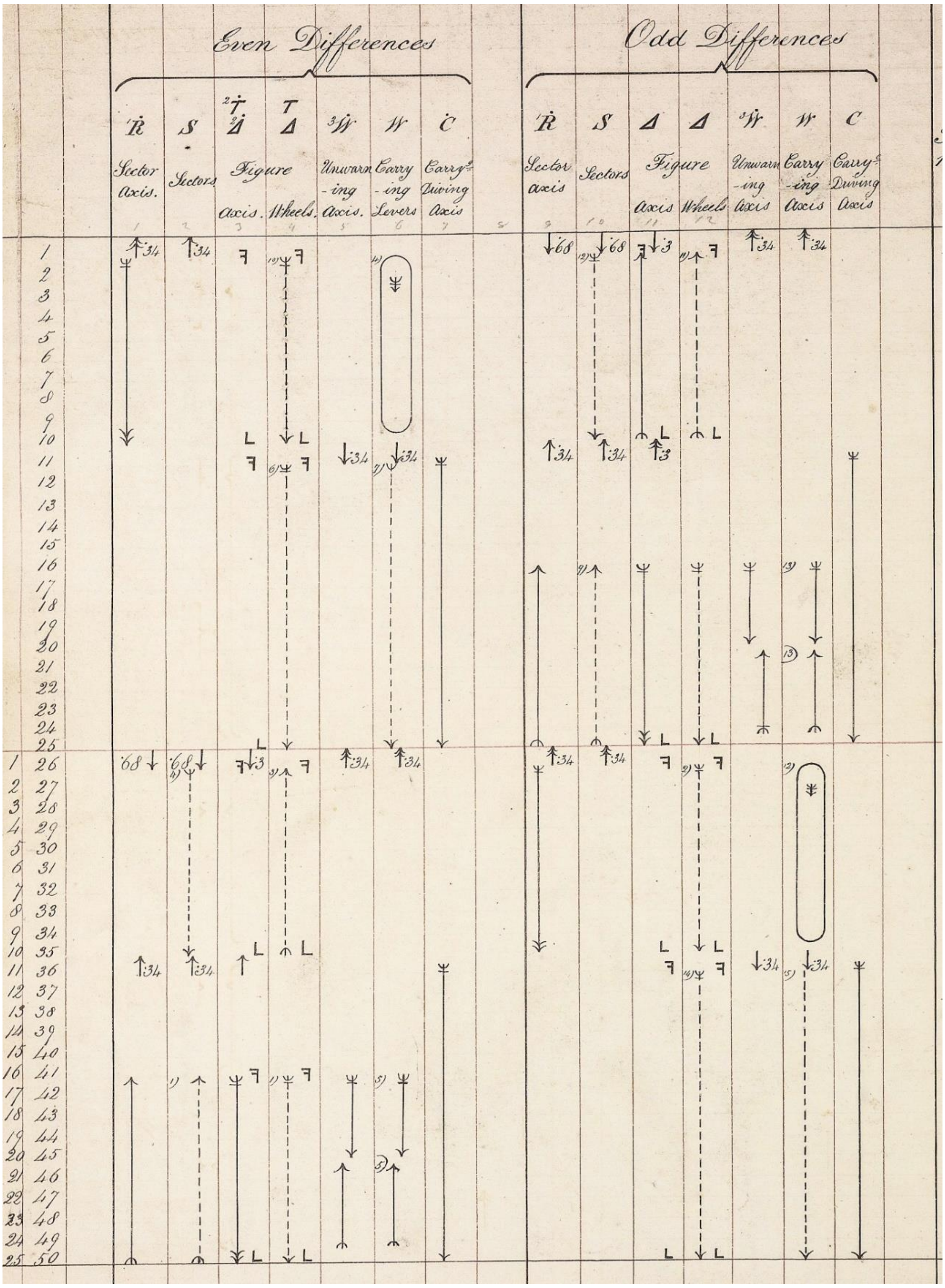


Fig. 3.2: Timing Diagram (F/385/1 (detail) (1848).

The symbols and graphics are part of the Mechanical Notation, the language of signs and symbols Babbage devised to describe, specify and optimise the designs (**Overview, Mechanical Notation**, pp. 6-9). In what follows the main use of the Notation is to identify parts using the lettered conventions as they appear in the drawings. Typically, the letters identifying parts have superscripts and subscripts. In this account the notations are used largely as uninterpreted labels to identify and refer to parts in the drawings. Sometimes, where of particular interest or where they have significance in the interpretation of the drawings, the notations are explained. The Mechanical Notation as a symbolic descriptive language is the subject of a separate study. Only a brief outline is given here.

In relation to the Timing Diagram (Fig. 3.2), the length of an arrow indicates duration of motion, except in the case of short vertical arrows with no tail symbols. The short arrows indicate lifting and lowering motions with the numbers alongside specifying the amount of physical displacement in inches. Tail- and head-variants of arrows indicate direction and nature of motion (circular/linear, clockwise/anticlockwise, and whether the motion returns the part to its rest position). A dotted arrow shaft indicates conditional motion – whether or not motion occurs throughout the time window of the arrow, only for part, or not at all. Vertical position on the page invariably indicates time of occurrence. The motion of locks is not represented in a separate column but is shown as a 'state' of a part, with a backwards 'F' (ꟻ) indicating 'Free' when the locks are disengaged, and an '**L**' indicating 'Locked'.

The timing cycle is tight and some of the critical transitions, especially when the locks engage and disengage, are too fine to be accurately specified with 50-unit resolution i.e. an event resolution of 7.2° was not fine enough to specify certain time-critical motions. Babbage's original Timing Diagram (which has omissions and inconsistencies) was refined to provide greater detail for the purposes of specification, mechanical design, and manufacture. It was also expanded to include the phasing of the locks as separate moveable parts as the timing of the operation of the locks has nearly no margin.

The fully detailed Timing Diagram (337 X 21) is the definitive reference source for the built Engine and frequent reference is made to it throughout this account (see p. 221).

Main Drawings: A/171, F/385/1, 337 X 21
Related Drawings: A/161, A/164, A/176, A/177

3.1 Giving-off

Tabulation using finite differences requires the repeated addition of multi-digit numbers. Drawing A/171 (Fig. 3.3) shows the mechanism for the non-destructive addition of a multidigit number on one column of figure wheels to a multidigit number on the figure wheel column to its immediate left.

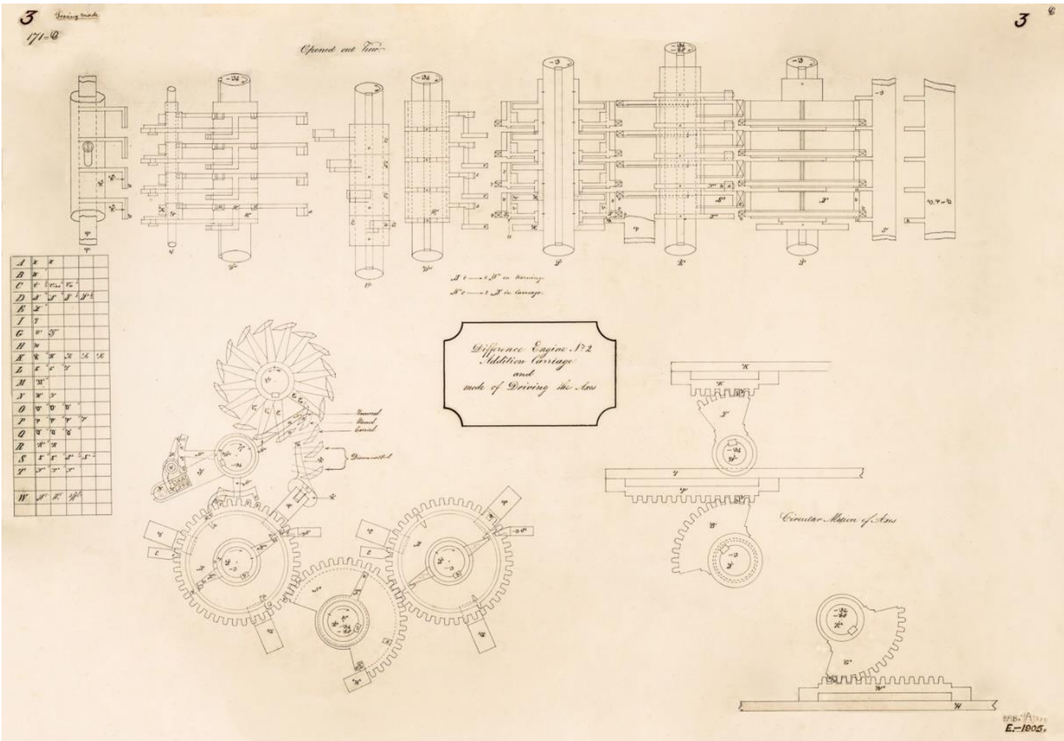


Fig. 3.3: Difference Engine No. 2, Addition and Carriage (A/171) (detail).

Drawing A/171 (Fig. 3.3) shows, in the view bottom left, two adjacent figure wheel columns ($n^{1}\boldsymbol{\Delta}^{2}$, $n^{1}\boldsymbol{\Delta}^{1}$) (expanded in Fig. 3.4). The cluster of parts at about 12-o'clock above the left-hand figure wheel (including the 'fairground' helix) is the mechanism for the carriage of tens. The wheel with the cut out (${}_{n}\boldsymbol{S}^{2}$) is a sector wheel in a column of sector wheels which interposes, wheel for wheel, between the two figure wheel columns.

The figure wheel on the right (${}^{2}\boldsymbol{\Delta}^{2}$) belongs to an even difference column, the one on the left (${}^{2}\boldsymbol{\Delta}^{1}$) to an odd difference column. The notations indicate that the figure wheels are those of the first difference (left) and second difference (right).

The view across the top is an elevation showing addition and carry mechanisms for a sample of four stacked figure wheels in each of the two difference columns. The mechanism has been opened out so as to expand the horizontal spacing for illustration. The 1st difference figure wheels are drawn in section to show the drive arms and internal nibs. The mechanism for the 2nd difference column (top right) is identical but the sectional detail of the figure wheels has not been repeated.

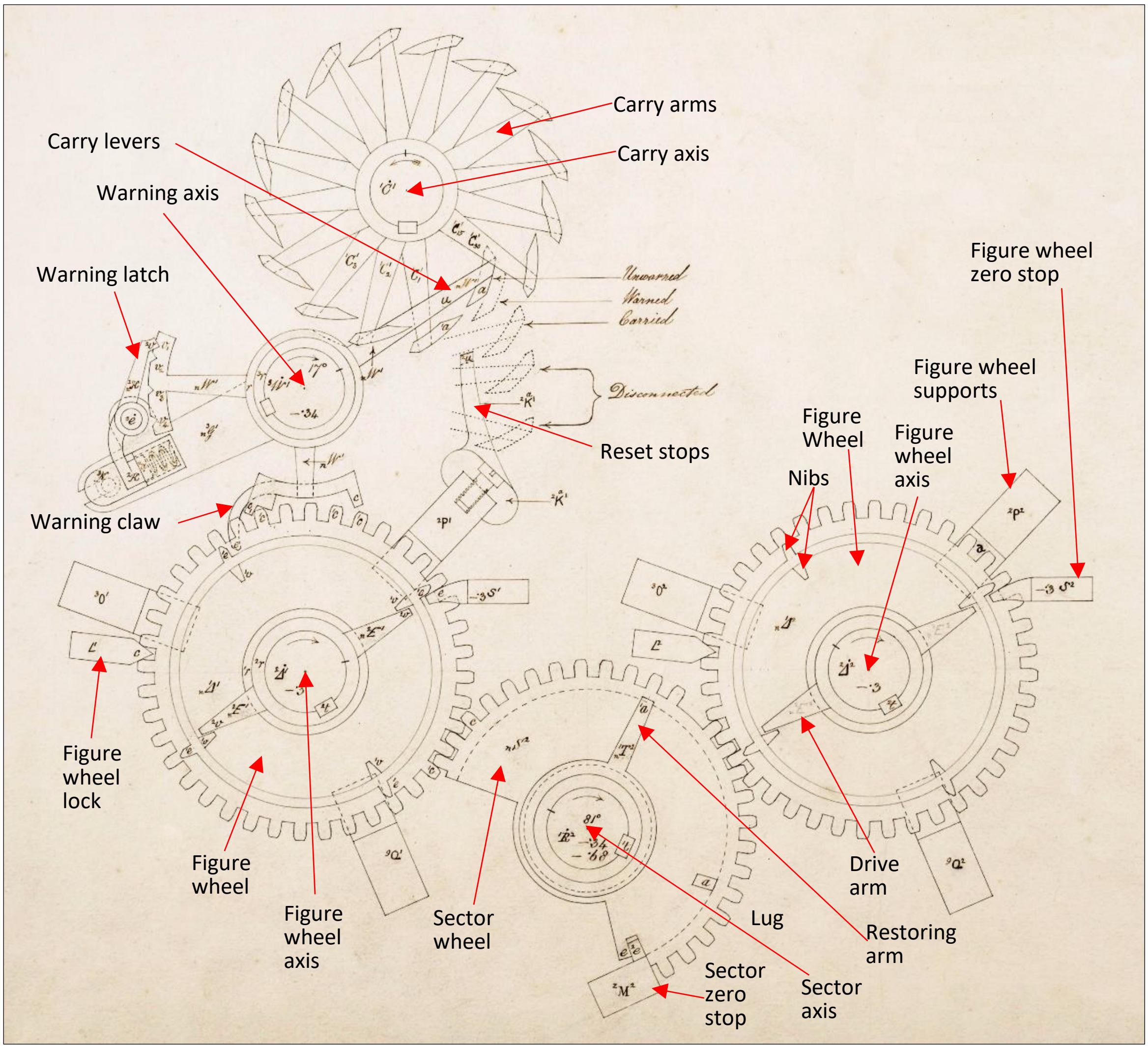


Fig. 3.4: Difference Engine No. 2, Addition and Carriage (A/171) (detail).

In the account that follows the vertical stacks of figure wheels and sectors are usually referred to as 'columns' when the context is their physical arrangement, and referred to as 'axes' when the context is the circular motions driven by the central drive shafts. 'Axis' often refers to both the figure wheels and the drive shaft. The two terms are interchangeable. The Intended meanings are almost always evident from the context.

Figure Wheels

The basic addition mechanism is shown in the sectional plan view (Fig. 3.4). The view features two figure wheels coupled by an intermediate sector wheel.

Each figure wheel has forty teeth with the pitch of one tooth (9°) corresponding to a single decimal number 0 to 9. Each wheel has four identical runs of decimal numbers and the same digit value can be represented by any one of four interchangeable angular positions referenced from an appropriate starting point. So each of the four quadrants of the figure wheel represents one decade 0 to 9 (Fig. 3.5).[1]

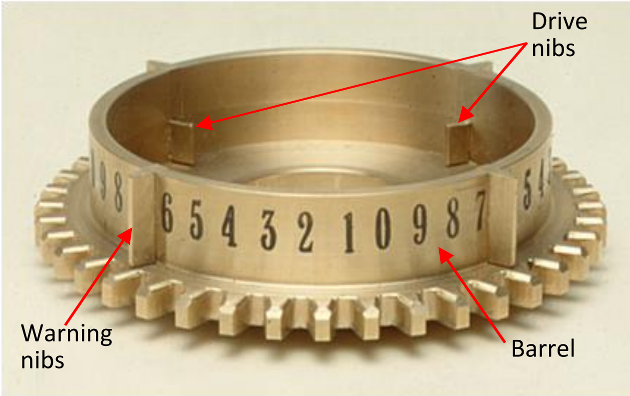


Fig. 3.5: Figure wheel. One of 248.

The figure wheels rotate freely on the figure wheel shafts i.e. they are not keyed or otherwise fixed to the shaft but rest on figure wheel supports. These are comb-like pillars that run the full vertical length of the columns with figure wheels resting on the equally spaced teeth of the comb (337 C 312). Each column of figure wheels is fitted with three figure wheel supports positioned roughly at 10-o'clock, 2-o'clock, and 5-o'clock, ([3]**O**, **[2]P**, **[9]Q**, Figs. 3.4, 3.6, 3.15). The figure wheel supports are part of the structural frame. The angular intervals between the supports around the figure wheel axis are not identical.

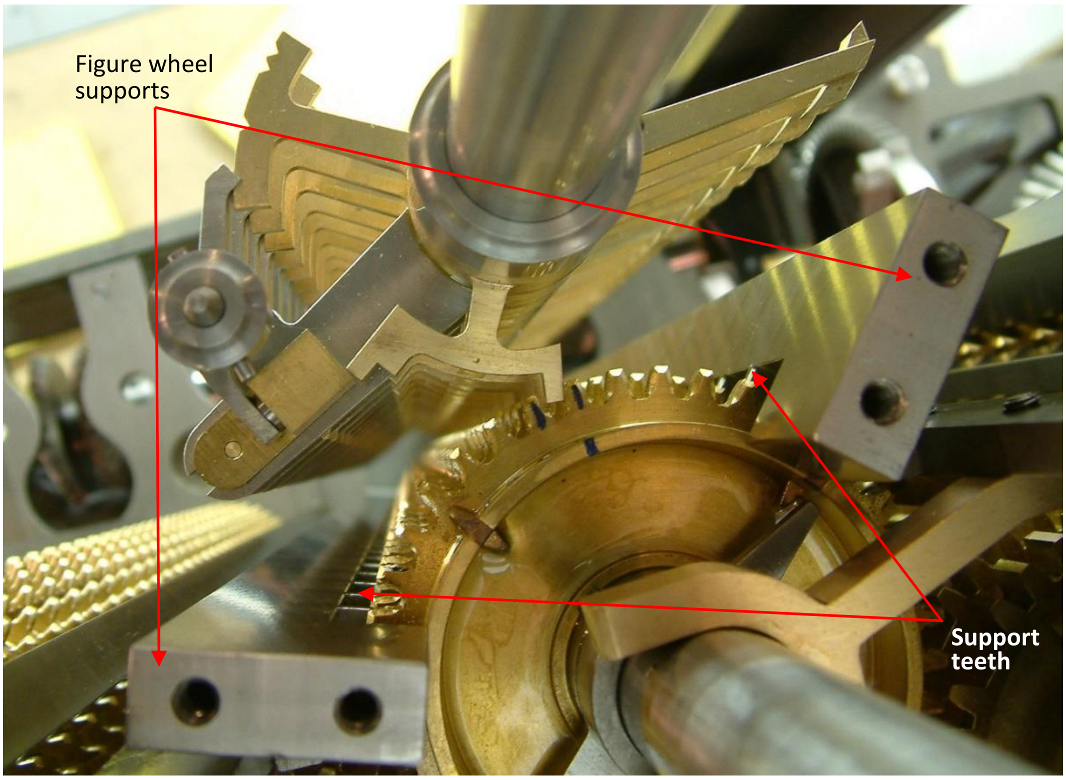


Fig. 3.6: Figure wheel supports (top plates removed).

[1] In the build Engine numbers on the odd figure wheels are engraved in reverse order to those on even figure wheels. This is because of a layout modification (see p. 38 and p. 47).

The figure wheels are driven by drive arms ($_{n}{}^{2}\boldsymbol{E}^{1}$, $_{n}{}^{2}\boldsymbol{E}^{2}$) keyed to the figure wheel drive shafts. The drive arms bear against internal nibs positioned at 90° intervals inside the barrel of each figure wheel i.e. there are four internal drive nibs inside each figure wheel barrel. The figure wheels themselves remain on the same horizontal plane throughout i.e. they do not move vertically but only rotate 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. Lowering the axis drops the drive arms into the plane of the internal nibs; lifting the axis disengages the drive arms by raising them clear of the nibs. The drive arms are disengaged when the wheels are being added to, and when manually setting the initial values at the start of a run of calculations.

Figure Wheel Zero Stops

As well as four internal nibs, each figure wheel has four external nibs integral with the barrel (Fig. 3.4, 3.5). The external nibs:

1. stop the figure wheels in the zero position and prevent over-running – this by running up against the zero stops ($\boldsymbol{S}^1$, $\boldsymbol{S}^2$ Fig. 3.4).
2. arm the carry warning mechanism by nudging the curved warning claw of the carry lever if the figure wheel passes through 9 (Fig. 3.4 left).

During each calculating cycle figure wheels are driven to zero by the internal drive arms bearing against the nibs on the inside of the figure wheel barrels. The zero stops halt the rotation to correctly position figure wheels when reduced to zero i.e. they prevent over-rotation past zero. The zero stops are not permanently fixed in the plane of the figure wheels. At appropriate points in the cycle they are raised to clear the external nibs so as not to obstruct the receiving wheel if its number value passes through 9 during giving-off.

The zero stops are lifted and lowered by the figure wheel axes to which they are coupled in a piggy-back arrangement (A/160 left view, right) (Fig. 3.7). A yolk ($^{7}\boldsymbol{K}^{1}$) at the base of the zero stop pillar ($\boldsymbol{S}^1$) engages with a collar ($^{2}\boldsymbol{B}^{1}$) on the lower part of the figure wheel axis. The yolk and collar arrangement allow the figure wheel axis to transmit vertical motions to the zero stops but not circular motions.

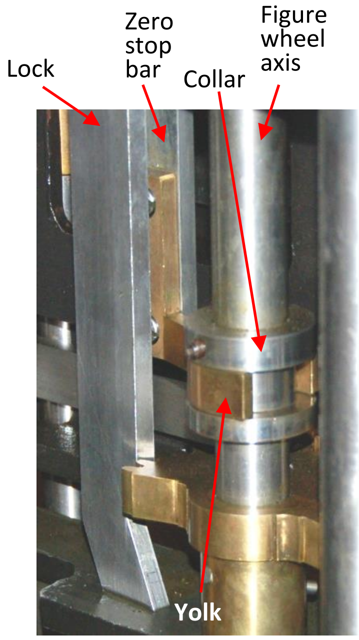


Fig. 3.7: Zero stop drive.

The vertical motion of the zero stop pillars is guided at the top end by bronze slider blocks on the top plates that span the topmost framing pieces. The tops of two zero stop bars are shown back to back in Fig. 3.8. This is a modification of Babbage's original layout. The revised layout is shown in 337 X 26 and discussed in **3.6 Resolution of the Layout Design Error**, p. 47.

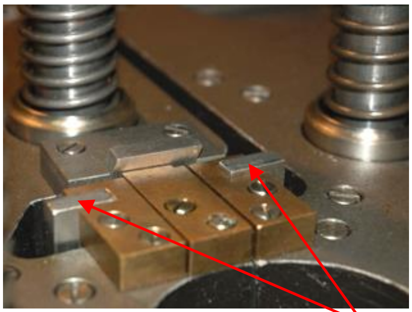


Fig. 3.8: Zero stop sliders.

Sector Wheels

The sector wheels, often called simply sectors:

1. couple and uncouple figure wheels in adjacent columns
2. restore the number given off so that this difference value is not lost and is available to be added to in the next calculating cycle.

The sector wheels (${}_{n}\boldsymbol{S}^2$ Fig. 3.4) are free to rotate on the sector axes i.e. they are not keyed to the shafts. They are driven by restoring arms (${}_{n}{}^{1}\boldsymbol{T}^2$) keyed to the shafts, which bear on a drive lug that protrudes above each sector (Figs. 3.9 centre, 3.10). The restoring arm drives the sector in the direction of the sector zero stop (${}^{2}\mathbf{M}^{2}$), reducing it to zero, and in doing so returns the sector to its rest position.

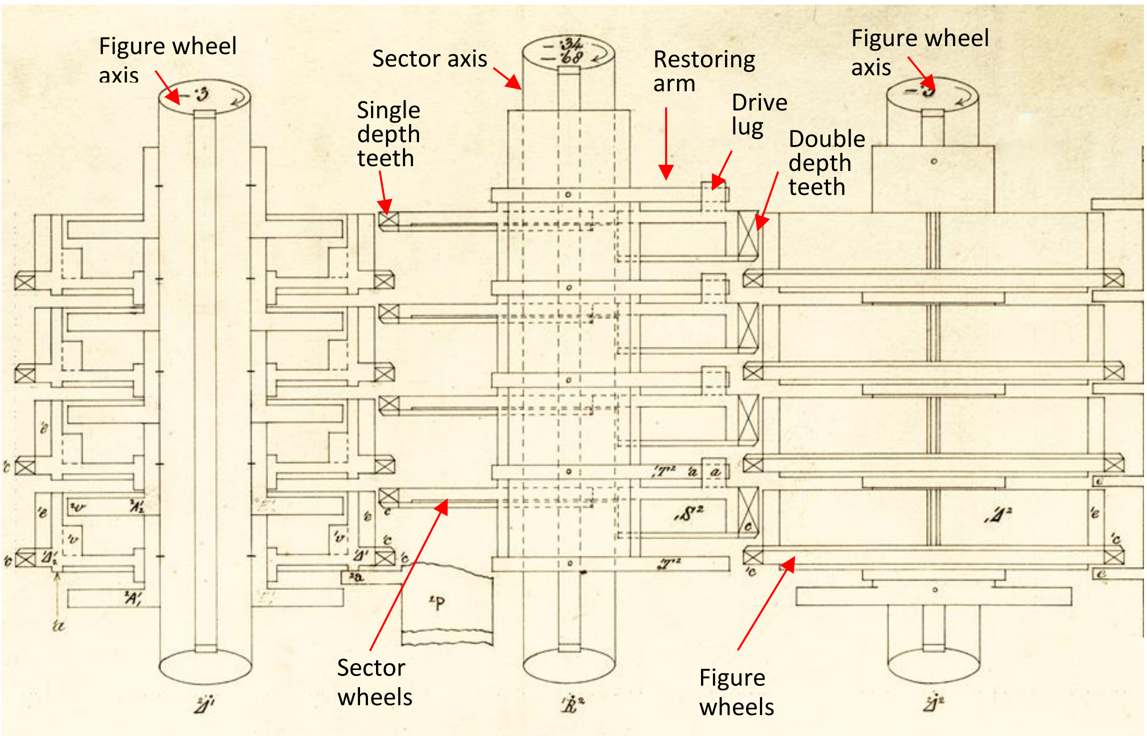


Fig. 3.9: Sector and figure wheels, Elevation (A/171) (detail).

Unlike the figure wheel drive-arms, which disengage from the internal nibs by being lifted clear, the sector drive arms are permanently fixed in the same plane as the drive lug. The restoring arms and sector wheels are lifted and lowered together with the sector axis in the same movement.

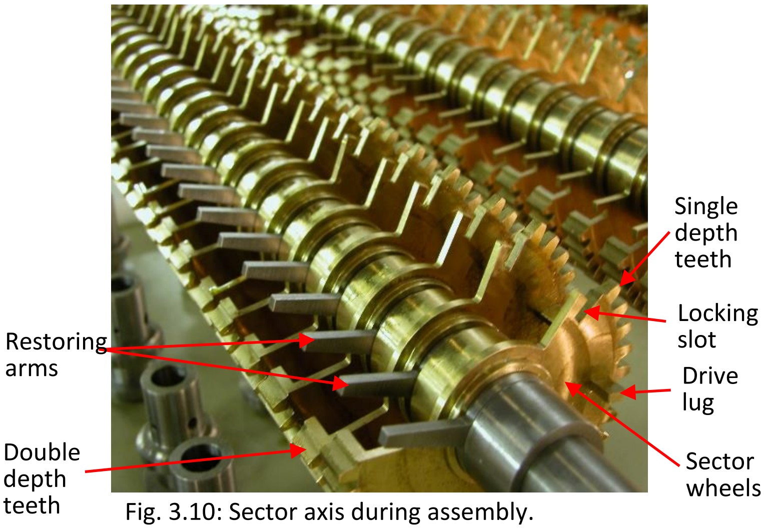


Fig. 3.10: Sector axis during assembly.

A sector wheel can mesh with one, both or neither of the two figure wheels on either side of it. When fully raised, the sectors are lifted clear of the figure wheels and disengage from both columns i.e. uncoupling the figure wheel columns from each other. This is the position shown in the opened-out view, Fig. 3.9.

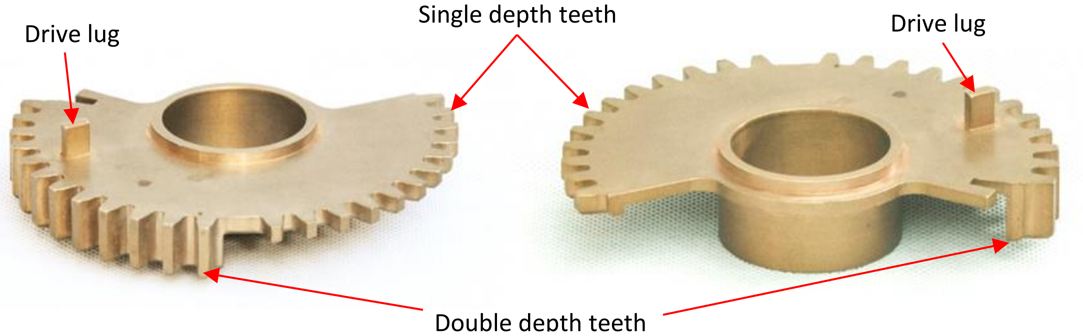


Fig. 3.11: Sector wheels: two views of same wheel.

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 run of sector teeth that mesh with the right-hand figure wheel is double depth (Figs. 3.11, 3.12). In the partially engaged position the sector can drive the right-hand figure wheel without affecting the left figure wheel. 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 wheel is transferred to the left-hand figure wheel, wheel for wheel and tooth for tooth, via the intermediate sectors as required during giving-off.

The rotation of the sectors is limited by zero stops ($^{2}\mathbf{M}^{2}$) which perform two separate functions:

1. stopping over-rotation of the sector by directly obstructing the path of the cut-away section of the wheel
2. locking the sector when fully disengaged i.e. in the fully raised position when the sector is restored to the rest position a lug on the zero stop engages with a slot in the sector wheel.

This locking action prevents derangement during the part of the cycle in which the sectors are not meshed with either figure wheel. The sector zero stops are not shown in the opened-out elevation in A/171 (Fig. 3.9).

Locks and Security Devices

Locking devices ensure the digital integrity of operation in which only discrete integers are legitimate representations of number values. The figure wheel locks are an example of the most the widely used form of locking mechanism ($\boldsymbol{L^1}$, $\boldsymbol{L^2}$ at about 9-o'clock, Fig. 3.4).

Figure wheel locks are long sword-like slats that run the full length of the figure wheel axis and act on all thirty-one figure wheels at the same time. The wedge-shaped edge is inserted between the figure wheel teeth at various points in the cycle.[2]

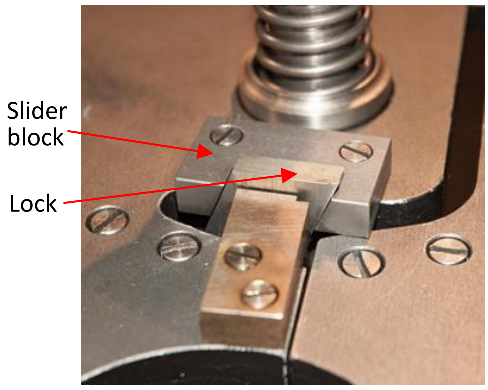


Fig. 3.12: Figure wheel lock and top slide block.

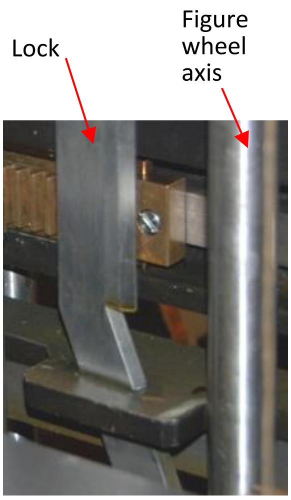


Fig. 3.13: Figure wheels lock and bottom plate.

The figure wheel locks are vertical at all times i.e. when they insert to engage the figure wheel teeth, the lock remains parallel to the figure wheel axis and moves uniformly in and out along its full length.

[2] Wedge locks are also used to align and lock the horizontal racks that interface the results column to the printer, the printing wheels, and drive pinions for vertical axes (see Chapter 4, **Output Apparatus.** Slotted locks are used to immobilise sectors in their rest position. Horn locks on the carry levers are used in the mechanism for the carriage of tens (see **3.2 Warning and Carriage of Tens**, p. 33).

The parallel lateral motion is produced by the lock being raised and lowered. The upper and lower ends of the locks are angled (Figs. 3.12, 3.13) and pass through angled slots in the framing, top and bottom. Raising the lock withdraws the wedge from between the teeth of the figure wheel and disengages the lock; lowering the lock drives the wedge towards the teeth. The locks are engaged and disengaged by the vertical motion drive at appropriate times in the timing cycle (337 X 21).

The operation of the locks is more visible in the Trial Piece[3] where they are not obscured by mechanisms alongside. The angled ends of the locks can be seen passing through the top and bottom plates. The locks on the Trial Piece are lifted and lowered by hand: lifting disengages the locks, lowering engages them.

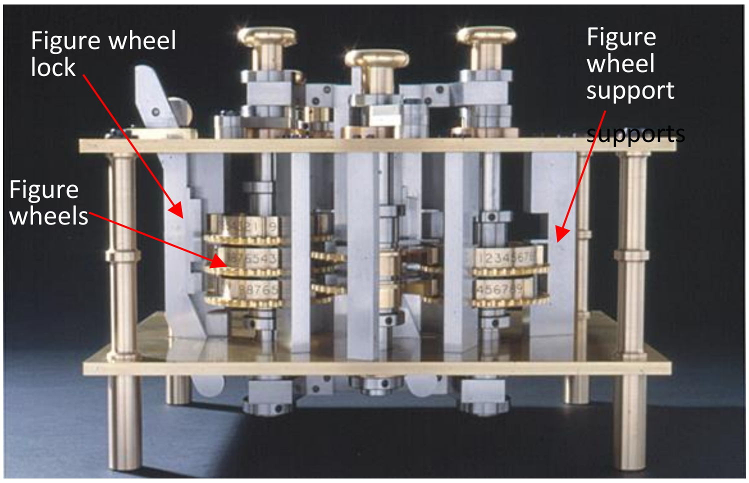


Fig. 3.14: Trial Piece.

The figure wheel locks perform three essential functions:

1. error prevention
2. error correction
3. error detection

With the lock engaged, the figure wheels are immobilised during intervals in the calculating cycle in which they might otherwise be vulnerable to movement not determined by deliberate mechanical control. The locking action is an error-prevention measure.

As the lock enters between gear teeth, the angled faces of the wedge true up the wheel and correct small derangements from exact integral values. This acts as a form error correction (analogous to 'pulse shaping' of the kind performed by a Schmidt trigger circuit in cleaning up the edges of a ragged electronic pulse). The centering action of the

[3] The Trial Piece was made before the construction of the full machine to verify and demonstrate the operation of the basic adding mechanism.

locks aligns the wheel to ensure that the position of a figure wheel coincides precisely with an integral tooth interval.

Finally, if a toothed wheel deranges by more than 2¼° (half the gap between adjacent teeth) insertion will be obstructed by the edge of a gear tooth. The lock will foul and the machine will jam. The crank handle seizes, and the operator is alerted that something is amiss. Jamming is not the misfortune it would ordinarily be but a form of error detection: a jam signals that the integrity of the calculation is compromised and that a wheel is in an intermediate position. The jam halts the calculation, signalling that a fault needs to be cleared.

Figure wheel locks are activated four times during each machine cycle to immobile figure wheels to lock their number values, correct small derangements, or polling for indeterminate positions (337 X 21 Timing Diagram).

The section of the figure wheel locks that extend below the stack of figure wheels also lock the circular motion quadrants (Fig. 3.7, A/171 lower right) that provide the rotational drive to the figure wheel axes (A/160 left, centre).

Initial Conditions

The basic addition unit is shown in Fig. 3.3 with the relevant part for giving-off repeated here (Fig. 3.15).

The Timing Diagram (Fig. 3.2) shows that the first addition is from odd to even differences and occurs in the first half-cycle, and from even to odd differences in the

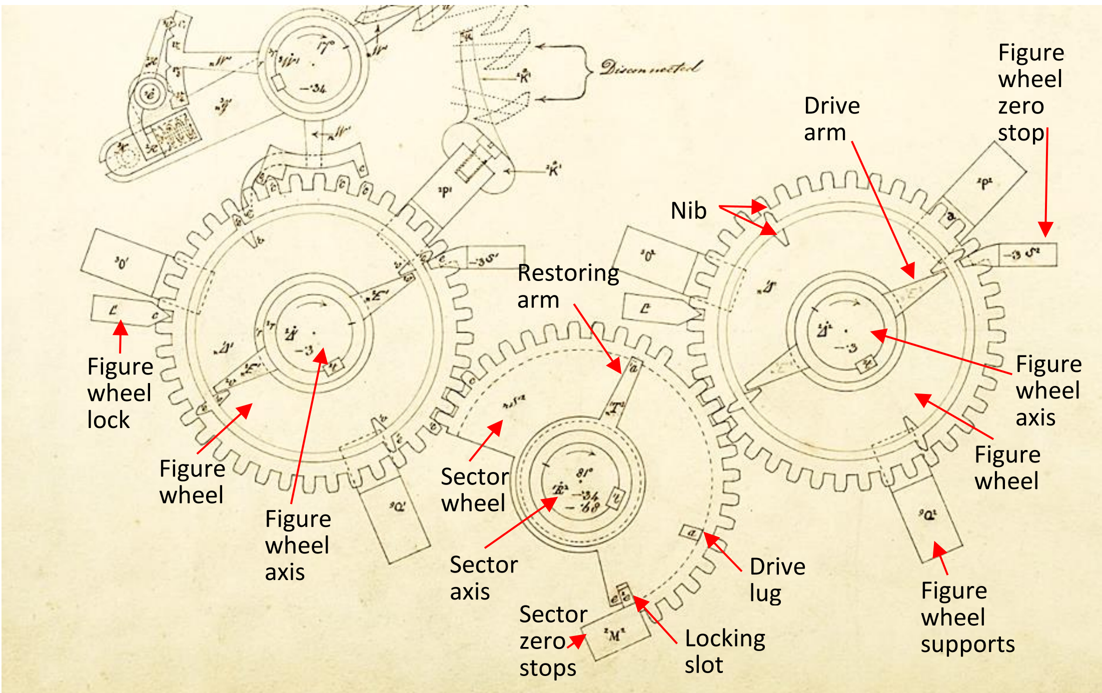


Fig. 3.15: Addition mechanism (A/171) (detail).

second half-cycle. However, the main drawing of the mechanism (Fig. 3.3, A/171) shows a layout for the addition of evens to odds. So that the description relates directly to the drawing, the account below starts with the second half-cycle i.e. half way through the Timing Diagrams – unit 25 in F/385/1 and 180° in 337 X 21.

The view shown in the main plan (Fig. 3.15) shows the position of parts at the start of the second half-cycle (180° in 337 X 21) during which the even difference on the right-hand figure wheel is added to an odd difference on the left-hand wheel. Both figure wheel drive arms are raised i.e. clear of the internal nibs. From the position of the zero stops in relation to the angular position of the external nibs, both figure wheels are shown set to zero. This is a special case and does not generally apply i.e. in general figure wheels will register a non-zero number at the start of the cycle and will be offset counter-clockwise from the zero position, by a variable number of teeth (9° per tooth) corresponding to the digit value for that number position. The internal nibs shown at 5-o'clock will therefore in general be somewhere in the quadrant between 5-o'clock and 2-o'clock i.e. in the most general case each of the thirty-one 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 the next addition half-cycle, zero (to thirty-one places) on the right-hand column will be added to zero (to thirty-one places) on the left-hand column.

At the start the evens-to-odds half-cycle (180° in the calculating cycle) the figure wheel locks are fully engaged and both columns are immobilised. The sectors are at zero i.e. fully clockwise against the zero stops and are fully raised (by the sector axis) and the restoring arms fully anticlockwise. With the sectors in the home position (i.e. against the zero stops) raising them locks them by engaging the locking lug on the zero stop with a slot in the sectors. In the fully raised position the sectors are disengaged from both sets of figure wheels so that the figure wheels are uncoupled from each other. The restoring arms are fully anticlockwise (at 1-o'clock). The sectors are against the zero stops at the start of cycle regardless of the numbers set on the right and left figure wheels at the start of cycle.

Operation

Giving-off starts with the sectors lowering 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 of the left-hand figure wheel to pass the zero-stop position without obstruction as is required if the figure wheel value exceeds 9 during giving-off.

The locks on both figure wheel columns lift to disengage 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 overshoot by blocking the path of the outer nib. The number held on the figure wheel at the start of the cycle (in the case shown in A/171 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 from the right-hand wheel completing the basic operation of addition. The right-hand figure wheel is now at zero and the original evens difference number is stored by the rotational displacement of the sector.

If during giving-off a left-hand figure wheel exceeds 9 then a carriage of tens is required to complete the addition. Tens carriage is not carried out during giving-off i.e. the action to increment the figure wheel above does not occur when the figure wheel passes from 9 to 0. Instead, during giving-off, a warning device is armed and latched to flag that a carry is still outstanding. The warnings are then serviced in the next phase of the cycle during which the tens are carried. Warning and carriage occur during separate parts of the cycle.

3.2 Warning and Carriage of Tens

The warning and carriage mechanism is shown in plan as a cluster assembly above the left-hand figure wheel in Figs. 3.3, 3.4) a detail of which is repeated below (Fig. 3.16). The mechanism consists of the warning axis (${}^{3}\mathscr{W}^{1}$**,** shown at 12-o'clock to the figure wheel axis), the three-limbed carry levers (${}_{n}\mathscr{W}^{1}$) mounted on the warning axis, the spring-loaded warning mechanisms with detent levers (${}_{n}{}^{1}\mathscr{K}$) mounted on the detent support arms (${}_{n}{}^{3}\mathscr{G}^{1}$) , the reset stops (${}^{2}\mathbf{K}^{1}$) which are fixed to the figure wheel support pillar shown at 2-o'clock to the figure wheel axis, and carry axis (${}^{1}\boldsymbol{C}^{1}$) with the helical arrangement of carry arms (${}^{1}\boldsymbol{C}^{1}{}_{n}$).

The carry lever is the most complex single part in the Engine. Each has three limbs which project radially from a central boss with which it is forms a single piece. The inset (Fig. 3.16) shows a loose carry lever inverted to show the warning claw detail. (In normal operation it is the other way up i.e. claw is on the underside).

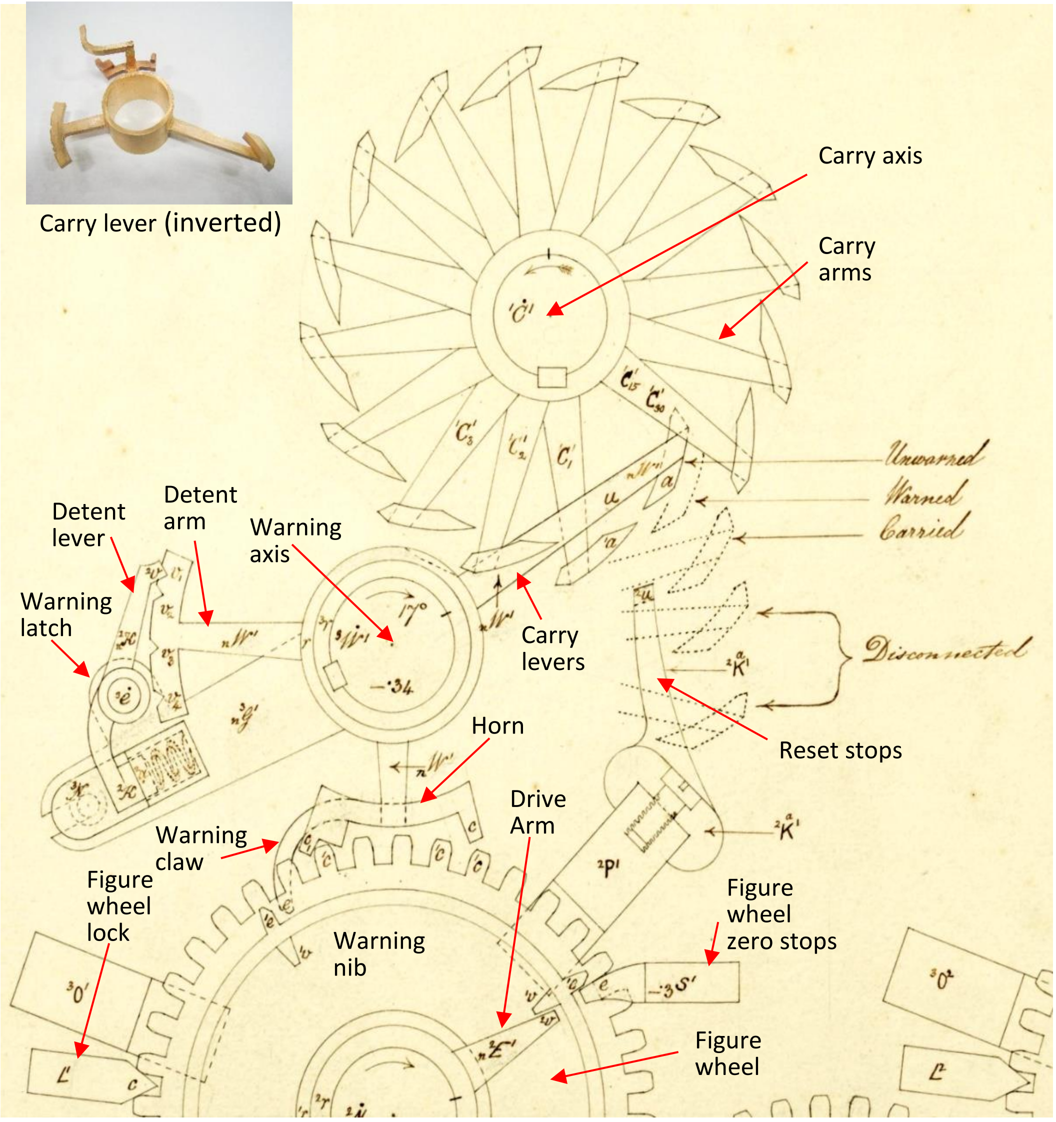


Fig. 3.16: Warning and Carriage Mechanism (A/171) (detail).

A carry lever performs three functions:

1. warns that there is a carry pending
2. locks the figure wheels to prevent derangement during the carry period
3. advances the next figure wheel above by one if the carry warning is set.

The carry levers do not pivot on the warning axis directly but are free to rotate on the steel boss attached to the detent support arm which is in turn keyed to the carry axis. In the mock-up in Fig. 3.17 the position of the figure wheel in relation to the carry lever is not exact – it is too far away from the figure wheel. The correct position is shown in Fig. 3.16 i.e. the cutaway horn (left) immediately above the claw is close enough to engage with the teeth of the figure wheel, and the right-hand horn should be poised to enter the gap between two figure wheel teeth immediately below.

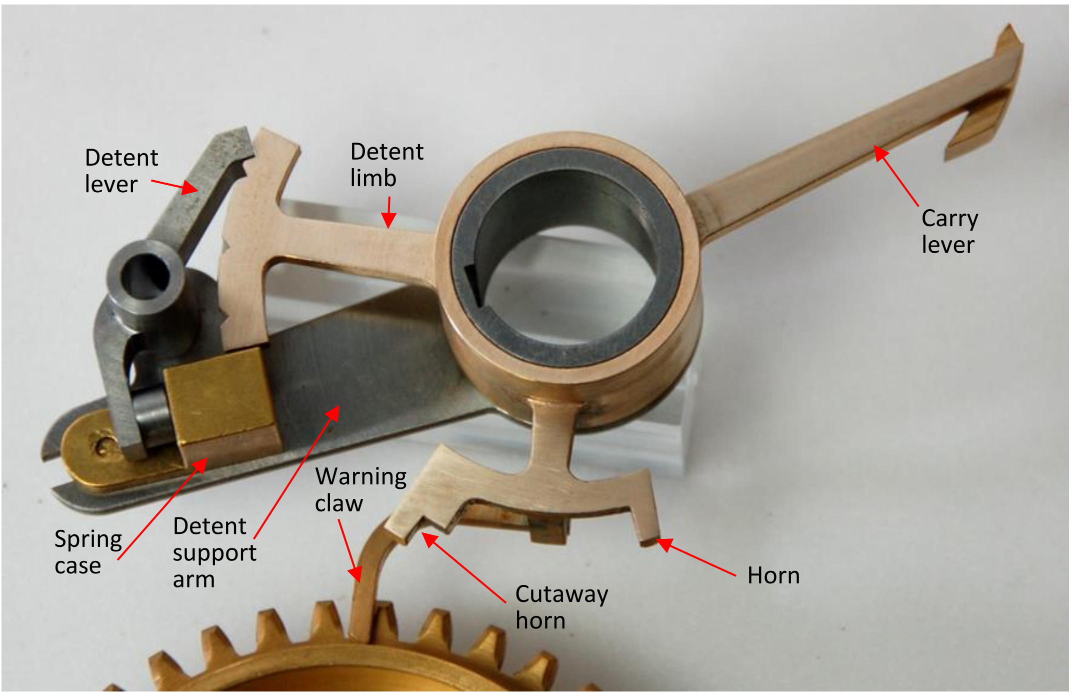


Fig. 3.17: Warning lever, latch and warning limb (placement of figure wheel is indicative).

The limb shown at 10 o'clock is the detent limb which has four positional notches in it. The steel detent lever, biased by a spring, captive in a spring housing fixed to the detent support arm, acts to latch the carry lever in one of four discrete positions determined by four V-notches (working points V_1, V_2, V_3, V_4 Fig. 3.16). The four angular positions of the lever correspond respectively to the four states, 'unwarned', 'warned', 'carried', and 'disconnected' (Fig. 3.16). The carry lever shown at 2-o'clock is drawn solid in the unwarned position (first V-notch) and as dotted partial views in the other three positions. (In the Notation V_1 to V_4 are 'working points' rather than parts identifiers. For ease of reference they are used here to identify the four V-notches.)

The lever shown just past 6-o'clock has two limbs: the curved warning claw, and two horns resembling an escapement. The claw and the horns are in different horizontal planes and span two figure wheel digit positions (inset Figs. 3.16, 3.19) i.e. the horns act on the figure wheel one above the one that operates the warning claw.

The horns have two functions: they prevent figure wheels from deranging while waiting for a possible carry from below and ensure that a figure wheel cannot be incremented unless driven from a legitimate source at the appropriate part of the calculating cycle (by a carry from below or as a result of giving-off from a higher order difference column alongside); the second function of the horns is to increment by one a figure wheel for the carriage of tens. These actions are described below.

Carry Arms

The helical 'fairground' arrangement in Fig. 3.16 consists of the carry axis (${}^{1}\boldsymbol{C}^{1}$) and carry arms (${}^{1}\boldsymbol{C}\,{}^{1}{}_{n}$). There are thirty carry arms with a fixed angular pitch of 22½° around the carry axis which creates the helical form, with a single 45° gap between the 15th and 16th carry arms as the only exception. The arms have the same fixed vertical pitch as the figure wheels so that each figure wheel has an associated carry lever in the same plane. The 45° gap between the 15th and 16th carry arms avoids fouling in two separate situations as explained below.


Fig. 3.18: Carry axis.

In the plan view each of the fifteen arms hides its counterpart in the run below i.e. $\boldsymbol{C}_{30}$ conceals $\boldsymbol{C}_{15}$, $\boldsymbol{C}_{29}$ conceals $\boldsymbol{C}_{14}$ etc. The uppermost carry arm (at 4-o'clock) is annotated '$\boldsymbol{C}_{15}$ $\boldsymbol{C}_{30}$' indicating that the single arm drawn represents the arm visible from above as well as the hidden arm below. Only the uppermost arm is so annotated (a drafting economy): a fully annotated drawing would have two notations for each arm.

Warning

As a figure wheel passes from 9 to 0 the external nib on the barrel of the figure wheel, in the same horizontal plane as the warning claw, pushes the claw outwards i.e. away from the figure wheel (Fig. 3.16). The passage of the nib past the claw nudges the carry lever clockwise from the unwarned position (first V-notch), to the warned position (second V-notch) where it is held by the sprung detent lever. The carry lever is latched by the detent lever in the warned position and, in so doing, storing the need for a carry until the next phase of the cycle. If the latch is set, the warning device is said to be armed. An armed latch signals that a carry to the next higher decade is required to complete the addition. The act of arming the warning latch, and the subsequent execution of the carriage, are separate actions i.e. the next wheel up is not incremented at the same time as the lower wheel exceeds 9.

Warning action occurs during giving-off and occurs only when a figure wheel value exceeds 9. Giving-off and warning occur within the same time window for all thirty-one digit-positions in the column. Warning occurs in general at different times for each figure wheel depending on the number value of a particular figure wheel in the course of a calculation.

During giving-off the warning axis carry levers are their raised positions (Fig. 3.19). The escapement horns of the carry levers are clear of the figure wheel teeth but the warning claw remains in the same plane as the external nib (Fig. 3.19). The rotation of the carry lever, nudged by the external nib, rotates the escapement horns so that the right-hand horn is poised above but clear of the gap between two figure wheel teeth below (Fig. 3.16).

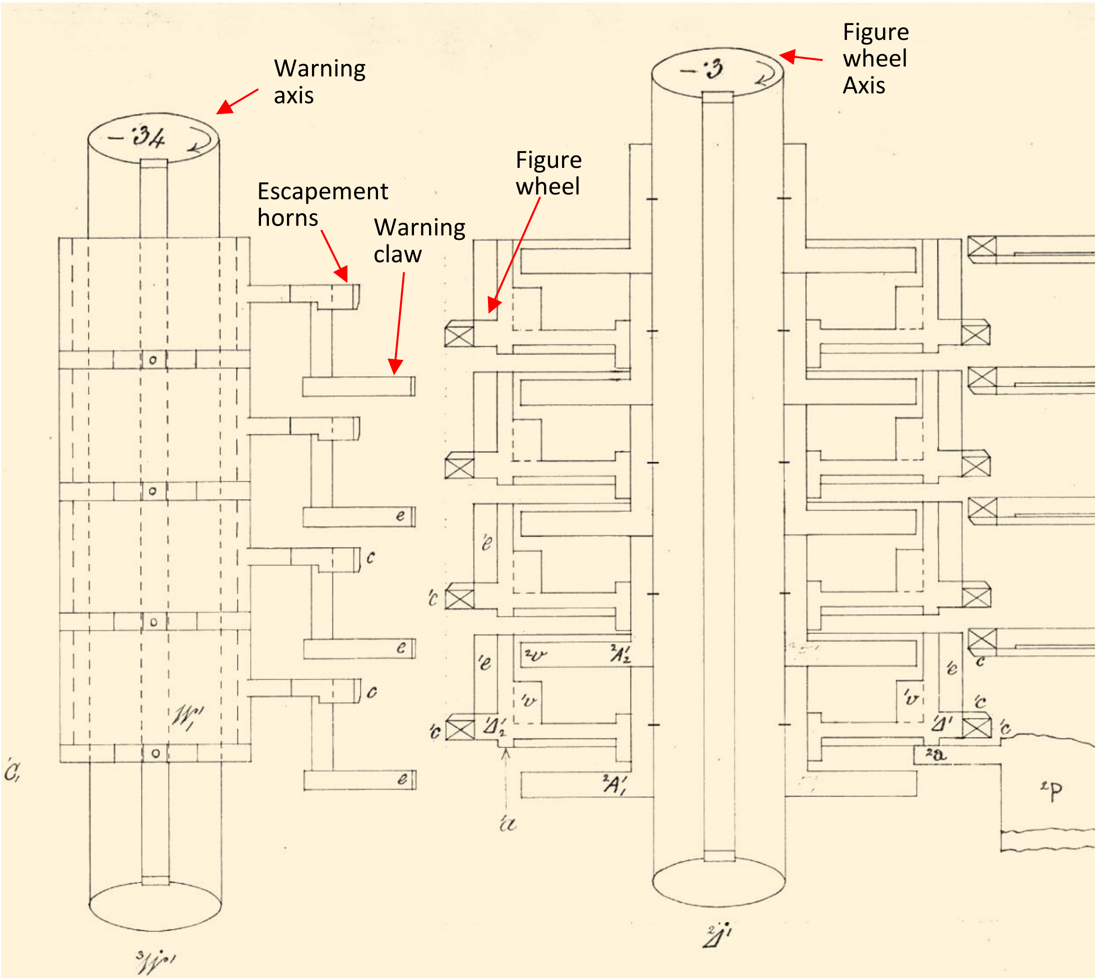


Fig. 3.19: Warning mechanism, Elevation (A/171) (detail).

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 (Fig. 3.15). At the start of the addition cycle the sector is in its home position against the zero stop with the restoring arm displaced anti-clockwise. The rotation of the sector during giving-off must therefore be counter-clockwise from the zero stop as this is the only degree of rotational freedom possible. It follows then that the right-hand figure wheels rotate 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. The warning mechanisms should therefore be armed by clockwise rotation of the outer nibs of the figure wheels during addition. However, with the warning mechanism as shown in Fig. 3.15 the warning claw will be correctly actioned by *anticlockwise* rotation of the figure wheel. 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 and the mechanism will not work if made as drawn.

The directional arrows on the figure wheel and sector axes in Fig. 3.15 (all of which are drawn with a clockwise sense) are little help. In the evolution of the Notation, arrow directions had different meanings at different times. At one point they indicated direction of physical rotation. Later this was changed to indicate 'positive direction' i.e. so as to increase number value. The idea of 'positive direction' was complicated by the sense of the numbering on the figure wheels i.e. direction of physical rotation to increase or decrease a number value depended on whether the numbers engraved on the figure wheels increased clockwise or counter-clockwise. The directional arrows in Fig. 3.15 are inconsistent however interpreted. The notations in the original Timing Diagram (Fig. 3.2) have their own omissions and inconsistencies. These too have so far defied coherent interpretation.

The principle of the mechanism is sound and it is possible that the inconsistency is no more than a drafting error. For the purposes of this description, we will continue to use the original layout – this so as to maintain Babbage's drawings as the primary interpretative source – but with the assumption that the left-hand figure wheel rotates anti-clockwise during giving-off. The problem and its solution are discussed more fully below (**3.6 Resolution of the Layout Design Error**, p. 47). In the meantime, a brief suspension of disbelief is required.

At the end of the warning phase the locks on both figure wheel axes engage. The left figure wheel lock engages and quickly disengages to correct minor derangements and to ensure that a figure wheel is not in an indeterminate position between two integers (Timing Diagrams 337 X 21, F/385/1, Fig. 3.2). The right-hand figure wheel lock engages and immobilises the figure wheels until they are again freed to move to have their initial values restored. At the same time as the locks engage, 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 so as to free the left-hand figure wheels for the carriage of tens (Fig. 3.19).

The carry levers are not locked and can be wilfully or inadvertently deranged to provide false warnings and false indication of carriage i.e. the levers can be moved by hand from unwarned to warned whenever the figure wheels are stationary, and from warned to carried at certain times in the cycle. Inadvertent derangement is discouraged by the sprung detent lever pressing into the V-notches but derangement is still physically possible.

Carriage

During the carry phase the warning mechanisms are serviced in turn and the carries propagate upwards i.e. the next figure wheel up is conditionally incremented by one tooth if the associated warning mechanism is armed. If the warning mechanism is not armed, no action to carry is taken.

The carry phase starts with the withdrawal of the locks from the left-hand figure wheels to free the wheels to receive a carry (337 X 21). At the same time the warning axis is lowered. This brings the escapement horns into the same plane as the upper figure wheel teeth. For unwarned figure wheels the cutaway horns on the left (Figs. 3.16) drop over and engage with the upper figure wheel. This locks the upper wheel to prevent derangement during the carry cycle when the figure wheel is otherwise unsecured. The right-hand horn is clear of the outer edges of the figure wheel teeth as shown in Fig. 3.16. In the case of warned figure wheels the rotational advance of the carry lever from ‘unwarned’ to ‘warned’ partially inserts the right-hand lug into the gap between two adjacent teeth ready to nudge the upper wheel one unit anticlockwise during the next phase of the cycle. The insertion of the lug has the additional function of acting as a partial lock.

With the carry levers in their lowered position the warning claw is still in the path of the external nib and unwarned carry levers can still be acted on by figure wheels that did not trigger warnings during giving-off. This allows for secondary carries i.e. carries that result from carries (see below).

Lowering the warning axis brings the carry levers into the same horizontal plane as the carry arms so that the lozenges of the warned limbs are now in the path of the lozenges of the corresponding carry arms (Fig. 3.16).

The rotation of the carry axis follows, and with it the rotation of the helical array of carry arms. With the angular stagger of the carry arms each figure wheel in a given stack is polled in turn starting from below as the carry axis rotates (in a technique Babbage called 'successive carry'). The direction of rotation is anticlockwise as indicated by the directional arrow drawn in the circle representing the carry shaft (${}^{1}\boldsymbol{C}^{1}$ Fig. 3.16). If a carry lever is unwarned then the trajectory of the carry arm lozenge does not intersect with the lozenge on the carry lever: the lozenges then pass without contact and no action to carry is taken. If the position is warned, the locus of the two lozenges intersect i.e. the act of warning during giving-off places the carry lever in the path of the carry arm. The outer face of the carry arm lozenge wipes past the inner face of the carry lever lozenge and pushes the carry lever aside as it slides past. This action of the carry arm nudges the carry lever one position on, from 'warned' to 'carried' (from $\boldsymbol{v}_2$ to $\boldsymbol{v}_3$) i.e. to the next discrete position on the run of V-notches. The clockwise rotation of the carry lever advances the right-hand horn of the escapement and this nudges the figure upper figure wheel one tooth on i.e. increments the number value by 1. 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.

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 arms 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 swirling carry arms are in different planes to the carry levers so do not interact. The inactive carry axis is captioned 'idling' in the revised Timing Diagram (337 X 21).

Secondary Carries

A primary carry is one that occurs as a result of a warning being set during giving-off. A secondary carry occurs when a figure wheel exceeds 9 as a result of a primary carry.

The most extreme case of a so-called 'domino carry' occurs when any non-zero number in the least significant digit position is given-off to a column with all figure wheels at 9. The lower-most (units) wheel is incremented 9 to 0 during giving-off which sets a warning for a primary carry. During the carry cycle the first carry arm services the warning and increments the next figure wheel which itself transitions from 9 to 0 setting the warning for the next figure wheel above. The newly armed warning mechanism is immediately polled by the next carry arm in the sequence and the secondary carry is propagated as before, and ripple carries can be propagated up the stack in this way. This successive carry operation 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.

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 15^{th} and 16^{th} arms is 45° not 22½° (gap shown at 5- o'clock in Fig. 3.16). The function of the gap is to prevent the carry arm and carry lever from fouling in either of two conditions. During giving-off the warning axis ($^{3}\mathscr{W}^{1}$, Fig. 3.16) is raised as shown in the elevation in Fig. 3.19. In this position the carry levers are not in the same plane as the carry arms and there is therefore no danger of the carry lever fouling a carry 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 16^{th} carry lever would foul the corresponding carry arm when the axis lowered. The double gap between the 15^{th} and 16^{th} arms allows the carry levers to lower without conflict.

The carry levers are only in the same plane as the carry arms only during the interval in which the carries occur (81°-168°, 261°-348°, 337 X21). Since, in normal operation, no primary warnings can occur in the carriage interval, there is no risk of rotational fouling i.e. the lozenge of the carry lever fouling the lozenge of a carry arm by attempting a warning were the double gap not there. However, if for whatever reason (a fracture in the drive train lifting the warning axes for example) the carry arms and carry levers were in the same plane and a warning was attempted the arms would foul. Again, the double gap removes this risk.

The carry axes complete two rotations for each carry action: the first rotation services carry warnings latched during giving-off in both upper and lower sections of the figure wheel axis; the second pass services a warning set during the first pass by the fifteenth figure wheel as this would not be cleared in the first pass. The double rotation ensures that this secondary carry, and any that result from it, are propagated i.e. that carries that result from carries are actioned.

The carry mechanism for individual figure wheels can be disabled by manually rotating the carry lever into the fourth detent position ('Disconnected' in Fig. 3.16) during the setting up procedure. Disabling the carry mechanism in this way prevents carries from propagating past the disabled figure wheel. This has the effect splitting the 31 digits into two and isolating the upper and lower sections of the figure wheel columns from each other. Separate and independent calculations can be run in the two sections and used for different purposes, as a cycle counter, for example, or to represent an automatically incremented argument in the tabulation for transfer to the output apparatus, or for a separate polynomial calculation carried out at the same time (see below). In the disconnected position the escapement horns of the carry-levers lock any unused figure wheels.

Moving the carry levers to the disconnected position is ordinarily blocked by the reset stops which are in the same horizontal plane. Disabling the carry levers requires raising the reset stops to allow the levers to pass under the stops. The stops are then lowered to trap the lever. The stops are freed for raising and lowering by releasing (by hand) fixing screws in slotted holes in the reset stop support pillar. The procedures are described in **User Manual (2013)**, 4.4 **Automatic Cycle Counting (**Step 5, p. 32).

3.3 Restoring the Value Given Off

Giving-off reduces the right-hand figure wheels (Fig. 3.16) to zero and the wheels no longer hold their initial values. The 'lost' difference value needs to be restored to the difference column for use in the next addition of differences.

During giving-off the sectors are displaced from zero by the initial value of the right-hand figure wheel (if non-zero). This value, stored on the sectors, is restored to the figure wheel during last phase of the second half-cycle.

At the start of the restoring phase the right-hand figure wheels are at zero and the sectors are displaced from zero by the initial value on the right-hand figure wheel (Timing

Diagrams F/385/1, 337 X 21). The sectors are fully lowered and are engaged with both left and right figure wheels (Fig. 3.9). Immediately after giving-off the sectors are raised to the partially engaged (middle) position i.e. meshed with the right-hand figure wheel but clear of the left wheel, and the right-hand figure wheels are locked and remain so until the start of the restore operation.

The locks disengage from the right-hand figure wheels and the sector restoring arms rotate in a clockwise sweep (Fig. 3.4) reducing the sectors to zero by bearing on the drive lugs. The sector zero stops prevent overshoot by obstructing the cutaway section of the sector. Since the sectors are coupled to the right-hand figure wheels, reduction to zero of the sectors transfers, tooth for tooth, to the figure wheel, the 'lost' initial value of the figure wheel. During restoration the sectors are uncoupled from the left-hand figure wheels and these remain unaffected by the rotation of the sectors. In reducing to zero the sectors restore the figure wheels values to those held before giving-off.

During restoration the right-hand figure wheel axis rotates anticlockwise and 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.

Restoration of the initial value of the right-hand figure wheel partially overlaps with the carry cycle of the left-hand figure wheels with both the carry and restoration phases ending at the same time i.e. at the end of the second 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 (337 X 21).

The end of the second 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, ready for the next half-cycle.

3.4 Resetting Warning Latches

There are two last actions needed to restore the mechanism to the start-of-cycle condition in preparation for the half-cycle that follows: returning the sector restoring arms to their anticlockwise home position; and resetting the armed warning mechanisms to their unwarned positions. Both these actions occur in the half-cycle that follows.

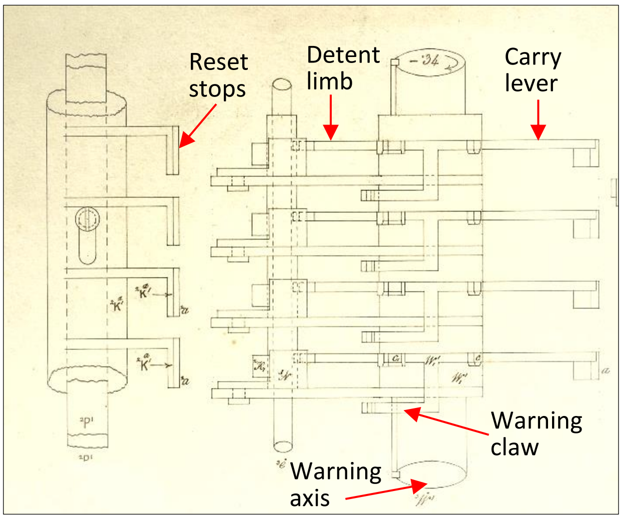


Fig. 3.20: Reset stops, Elevation (A/171) (detail).

In the configuration shown in Fig. 3.4 (evens to odds), resetting the odds warning mechanisms to the unwarned positions occurs at the end of the first half-cycle. The warning axis has already been raised so the escapement horns are above 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 (Fig. 3.16). The spring-loaded detent lever, mounted on the detent support arm, is carried clockwise, and the carry levers are driven towards the reset stops by the detent limb. The carry levers are in the same plane as the reset stops (Fig. 3.19) and the carry levers in the carried position will be driven against the stops and their motion halted. The detent support arm continues clockwise, and the detent lever is driven in turn from the carried V-notch ($\boldsymbol{V}_3$) to the warned notch ($\boldsymbol{V}_2$) and finally to the unwarned notch ($\boldsymbol{V}_1$) as it rides along the notched outer curve of the detent limb. The unwarned carry levers are unaffected: they are already reset and do not bear on the reset stops.

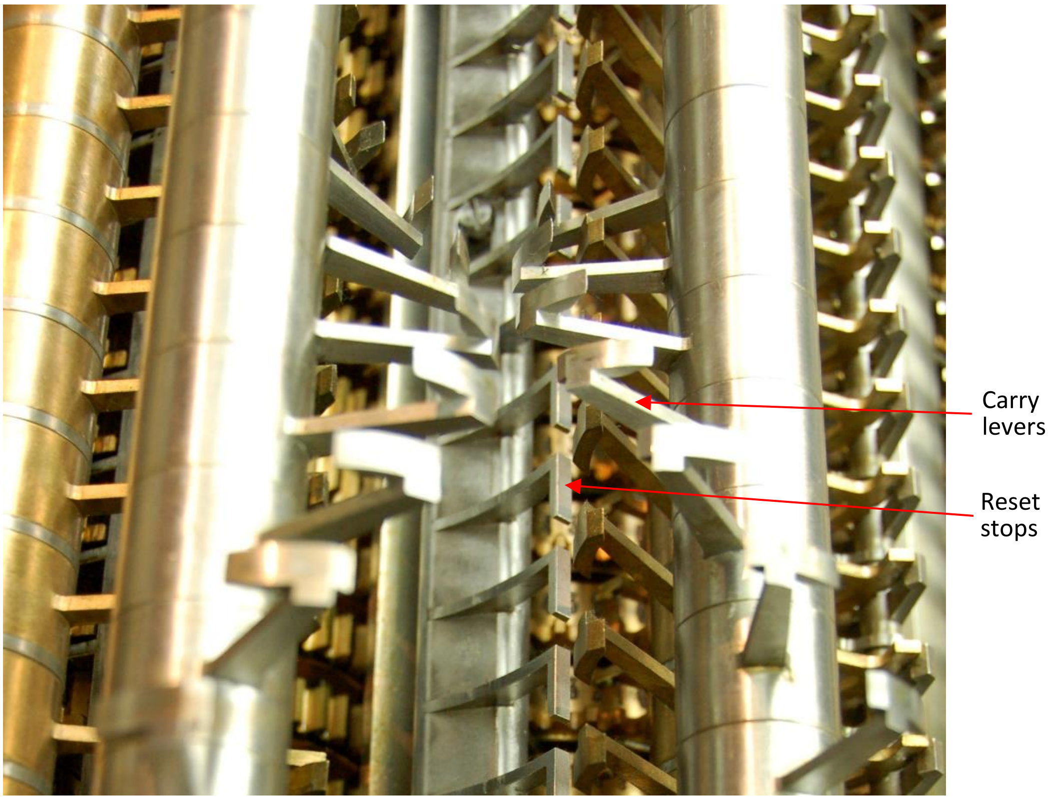


Fig. 3.21: Reset stops and carry levers.

The clockwise rotation of the detent support arm is followed immediately by a counter clockwise return motion which returns the whole assembly, with all levers unwarned, to their home positions (Fig. 3.16). The carry levers in the ‘Disconnected’ state are unaffected by the warning reset: disconnected levers are trapped behind the reset stops and, during the movement of the detent support arm, simply wave back and forth between the two disconnected positions (Fig. 3.16). The clockwise/anticlockwise waving motion of the warning axes to drive the carry levers against the reset stops, 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 (Fig. 3.2).

In the operations described above the carry lever plays a critical multifunctional role. Motion is imparted to the carry lever in four distinct ways for four separate operations: warning, carriage, reset and when disabled i.e.

1. by the outer nib of the figure wheel acting on the warning claw as the wheel is driven past 9 during addition. This advances the lever from ‘unwarned‘ to the ‘warned’.
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 and reset stops when the mechanism is reset after each addition cycle. The reset function returns the carry lever from ‘carried’ to ‘unwarned’ (first V-notch).
4. by hand to the ‘Disconnected’ position. This disables the transmission of carries.

3.5 ‘Pipelining’

In the pen-and-paper use of the method of differences the higher order difference is added to the next lower order difference, the lower order difference is then added to the next lower order difference and this process is repeated serially to generate each next tabular value. In the Engine the progression from the highest difference column through to the tabular value does not occur in a stepwise sequence right to left from column to column in seven distinct repetitions of the same sequence. Instead the calculating cycle is split into two halves. The four odd difference columns are added to the even difference columns in the first half-cycle, and all the even difference columns are added to the odd difference columns in the second half-cycle. (The tabular value column is classed as an even column). The overlapping concurrent operations are analogous to ‘pipelining’ in modern computing.

Provided this phasing arrangement is taken account of when setting up at the start of a run, by offsetting the initial values on alternate columns, the end result is the same. Carries are executed within each half-cycle so that the additions in each half-cycle are complete. In this arrangement a single calculating cycle consists of two half-cycles with odds-to-evens addition in the first half-cycle and, in a repetition of the sequences described above, evens-to-odds in the second half-cycle. Each full cycle, consisting of two half-cycles produces a new tabular value, and cycling the Engine produces the tabular series with each new tabular value appearing on the results column after each odds-to-evens addition.

The phasing of the two half-cycles and the interleaving of the various actions are shown in the two timing diagrams (F/385/1, 337 X 21).

'Pipelining' has several advantages:

1. **Time Efficiency**: if the addition (with carry) of two multi-digit numbers takes say 1 unit of time then seven serial additions take 7 units. The pipelined arrangement generates each tabular value in 2 units of time and so in the case of a 7^{th} order polynomial offers a 7:2 speed advantage.

2. **Machine Usage:** adding each difference to its neighbour in separate serial operations involves one column pair at a time i.e. only two of the eight columns are active for each addition and 75% of the engine is idle. In the pipelined arrangement all the columns participating in the calculation are active throughout the tabulation.

3. **Scalability:** with pipelining the cycle time of the engine is independent of the number of differences used in the calculation and the Engine could be extended to a greater number of differences without time penalty.

4. **Simplicity:** the parallel addition of odds-to-evens and evens-to-odds in the pipelined arrangement allows for a simpler control mechanism than required for a serial addition sequence. Adding column pairs serially requires a control system that independently activated each column pair in turn – a substantially more complex design than required for the phased pairing of alternate axes at the same time

3.6 Resolution of the Layout Design Error

The analysis of the warning mechanism as drawn flagged that the clockwise rotation of the left hand (odds) figure wheel (Fig. 3.3) was inconsistent with correct warning action and that and that the mechanism will not work if made as drawn. Put differently, during giving-off, the left-hand figure wheel needs to rotate anti-clockwise to correctly arm the warning latch, yet from the layout it is clear that the figure wheel rotates clockwise and the nib on the figure wheel will foul the warning claw and act as a stop.

It is unclear whether this conflict is a design oversight, a drafting error, or a deliberate error to foil plagiarism or espionage. While the error is a fundamental one it does not compromise the viability of the design, but the mechanism as drawn will not function as intended.

Several solutions were considered. The one chosen involves mirror-imaging the layout of alternate axes i.e. opposite-handing the odd and even axes. This retains the form of all parts but alternate axes are opposite handed. The revised layout is shown in Fig. 3.22.

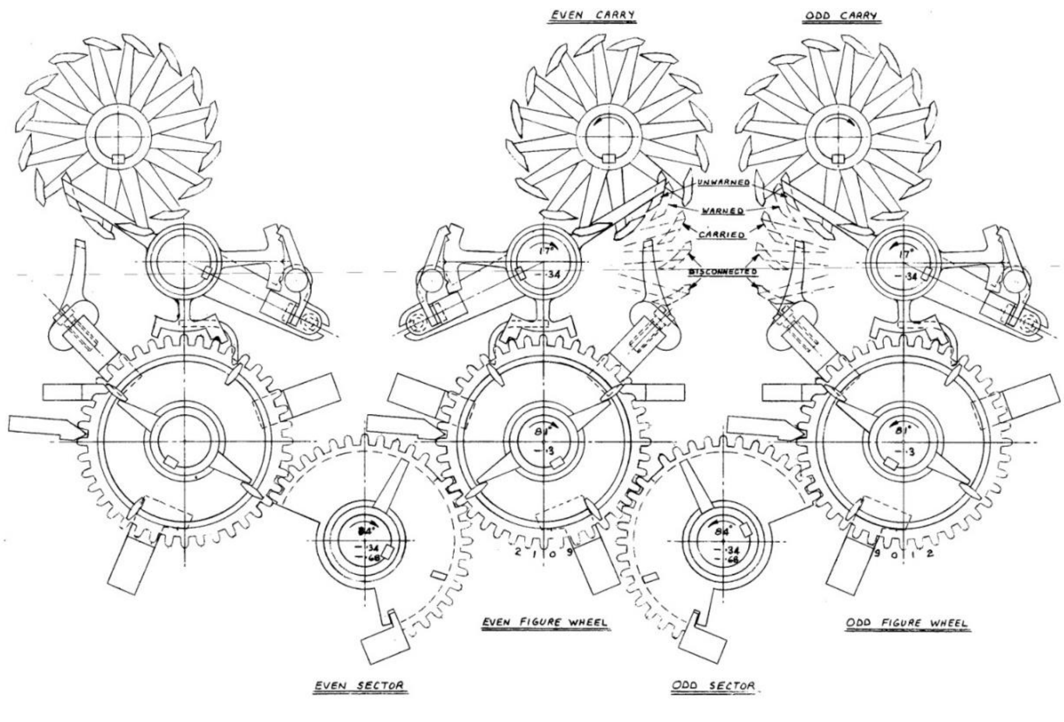


Fig. 3.22: Revised layout showing handing of alternate axes

The relationship of the axes for even differences is handed to the right as in the original layout. The relationship of the axes for odd differences is mirrored and handed to the left. This involves most parts on the odd difference axes being opposite handed: carry levers, carry lever support arms, carry arms, direction of numbering on figure wheel barrels, sectors, reset stops, sector and figure wheel zero stops and so on. The figure wheel locks are not mirrored – this to maintain the uniform pitch of the bell cranks that lift and lower the locks. (see **7. Drive, Lifting and Lowering**, pp. 175-178).

The layouts shown in related drawings A/161, A/164, A/176 and A/177 indicate that the two-column mechanism shown in A/171 is a generic unit identically repeated for the remaining four column pairs i.e. the layout of the axes and the orientation of the carriage mechanism is the same for odd and even differences throughout the machine. Mirroring alternate axes sacrifices this uniformity.

Mirroring alternate axes involves small modifications to the position of the various elements of the mechanism around the axes. The gear wheels in Fig. 3.3 are not drawn symmetrically around a centre-line between the axes and to preserve symmetry of form and timing, the mirrored arrangement requires repositioning parts around the axes by 2¼°, and carrying this adjustment through the whole specification. The mirrored arrangement also requires the reversal of the direction of rotation for certain circular motions. These modifications and adjustments are described in what follows as they arise.

Unequal spacing between figure wheel columns

There is a less serious layout anomaly that was more easily resolved. The main elevation of the Engine (A/163, Fig 3.1) shows a disruption in the standard pitch of the figure wheel columns. Specifically, the space between the fifth and sixth difference columns is shown larger than that between any other column pair. This exception is specifically repeated in A/164 though masked in A/161, A/176 and A/177 where the full run of 8 columns is not shown but indicated by broken lines implying repetition. No significance could be found for this anomalous gap and this was assumed to be a drafting layout error. This conclusion is confirmed by a tracing BAB/B/002 of the full plan view A/164. The tracing shows the corrected spacings between the columns as equal.

4. Output Apparatus

Integral to the concept of the Difference Engine is the automatic production of printed mathematical tables by mechanical means. Babbage's intention was that the 'unerring certainty of mechanical agency'[1] would eliminate the risk of human error in calculation, manual transcription of results, conventional typesetting using loose type, and proof reading.

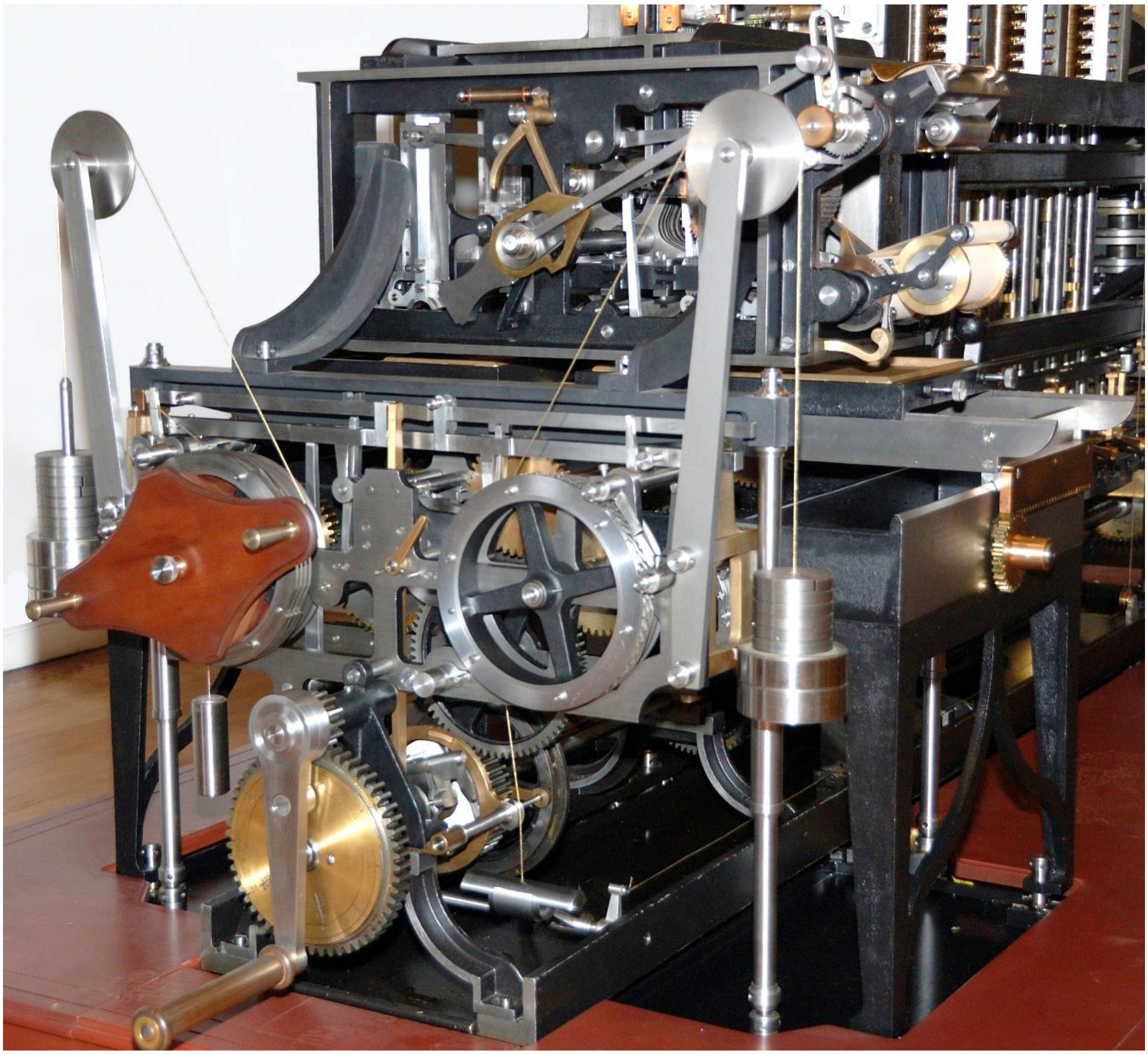


Fig. 4.1: Output Apparatus, Difference Engine No. 2.

[1] Lardner, Dionysus. "Babbage's Calculating Engine." *Edinburgh Review* 59 (1834): 263-327. Reprinted: "Babbage's Calculating Engine." *The Works of Charles Babbage*. Ed. Campbell-Kelly, Martin. Vol. 2. London: William Pickering, 1989. Pp. 118-187, p. 169.

The output apparatus prints tabular results in inked hardcopy on a paper print roll, and impresses results into soft material held in trays. The impressions were intended to serve as moulds for the production of printing plates for use in a conventional printing press in a process known as stereotyping.

Tabular values from the results figure wheel column are automatically transferred to the output apparatus. There is no buffering or storage of results. Each result is printed and stereotyped during the calculating cycle that generates it.

The output apparatus consists of two sections, the printing apparatus that produces the inked hardcopy, and the stereotyping apparatus for the production of the stereotyping moulds (Figs. 2.2. 2.3, 2.5, 4.1). The whole apparatus, that is both sections, is often referred to as the printing mechanism, or printer. Though the apparatus does produce a printed paper record, the primary function of the apparatus is stereotyping i.e. impressing results on soft material, wet plaster, for example, to produce a stereotype mould from which printing plates could be cast for duplication of results by conventional printing which was the principal method of dissemination (Figs. 2.6, 2.7). The inked printout is one-off and used for checking purposes and as a record of the contents of the stereotypes.

The stereotyping table (lower section left, Fig. 2.3) is larger front-to-back than the calculating section and provides the platform for the stereotyping trays that receive the results.
Results are impressed in two trays of different sizes at the same time in two different fonts, small and large (Fig. 4.3). The stereotyping punch wheels lift and lower to impress results but do not otherwise move along or down the page. Instead the moving platform automatically repositions the trays under the print heads to receive each new result. The platform and trays move in the X-Y plane only i.e. in the horizontal plane; the punch wheels move in the vertical plane.

The format of results impressed in the trays is ‘programmable’. Combinations of layout features can be selected including the number of columns per page, whether results appear in order down the page (line-to-line with automatic line-wrap) or column-to-column across the page (with automatic column wrap), line height, leaving blank lines between groups of lines for ease of reading, page and column margins. Formatting options are chosen by selecting (by moving a sliding pawl) two from eight pattern wheels where the pitch of the teeth and their arrangement on the wheel determine the layout options. One set of four pattern wheels controls line-to-line behaviour, the other set of four, column-to-column behaviour (Fig. 2.5). Individual layout features are not independently variable but come in fixed combinations with each pattern wheel determining one set of combinations. Any of the pattern wheels can be swapped out with pattern wheels customised for combinations other than those provided as standard.

Formatting options are available only for stereotyped results impressed in the trays. Inked hardcopy printed on the paper roll is in a fixed and unalterable format with the line height, character spacing, and font size fixed and with results printed in a single column, one result per printed line (Fig. 2.6).

The operations of the inking, printing and stereotyping mechanisms are co-ordinated and controlled by a set of vertical cams located inside the frame for the printing apparatus frame and driven, via gears, from the main drive shaft on the underside of the Engine (Fig. 2.1). The cam cluster consists of fourteen cams of which ten are conjugate pairs i.e. two cams that are geometric inversions of each other – where there is a rise on one, there is a corresponding fall on the other so that the cam followers provide positive drive in both directions.

A second feature of the control mechanisms is the use of locks that operate in five separate places in the output apparatus: the function of locking and security devices is to preserve digital integrity of operation. Various locking methods are featured throughout the Engine the most widely used of which is the wedge lock. A wedge-shaped bar or slat is inserted between gear teeth or V-shaped notches at specific points in the cycle. The insertion of the lock immobilises the component so as to prevent derangement (error prevention), makes minor corrective adjustments to centre the component in the event of small derangements or slightly inexact resting positions (error correction) or, in the event that a component is displaced outside the valid operating range, the entry of the lock can be obstructed and the machine will jam, indicating that the integrity of operation has been compromised (error detection).

All five of the locks in the output apparatus are wedge locks. These were implemented as originally drawn. It was found that while the printed results were correct, the alignment of type was inexact. In this case the force of entry of the locks was increased to improve the centering action beyond minor correction, to ensure alignment of the type to analogue standards of exactness comparable to manual typesetting using frames.

The output apparatus is bolted directly to the frame of the calculating section to which it is coupled as shown on the left in the main elevation drawing (Fig. 2.3). The stereotyping table is supported by its own cast stand which arches over and straddles the two long rails of the engine base.

Only the thirty least significant digits of the 31-digit result are transferred from the calculating section to the output apparatus: the most significant digit, represented by the uppermost figure wheel on the tabular column, is not transferred or printed. This digit serves a guard digit and as an overflow warning.

There is no printing overhead in the timing cycle: stereotyping and printing do not lengthen the timing cycle but take place as parallel actions during the standard calculation cycle.

The general arrangement is shown in Fig. 4.2. The illustration shows the mechanism for transferring two digits of a thirty-one-digit result from the tabular figure wheel column of the calculating section to the printing and stereotyping apparatus. The mechanisms are identical for omitted digits.

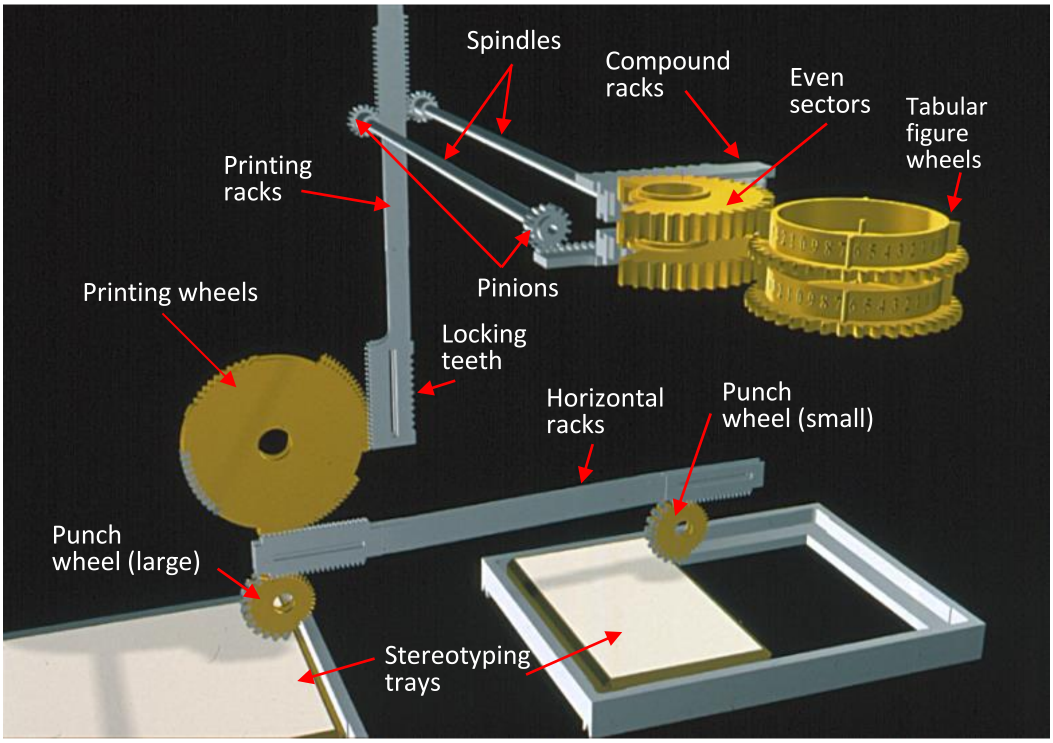


Fig. 4.2: Transfer of results to output apparatus (simulation).

Results are transferred from the tabular figure wheels (the last even figure wheel column, far left, Fig. 2.3) to the vertical racks (called ‘Printing Racks’ in A/172, top centre) by a transmission train consisting in turn of tabular figure wheels, sectors, compound racks, pinions and spindles. At the far end of the spindles is a set of vertical racks and pinions. (The compound racks are called ‘Figure Racks’ in the full Timing Diagram (F/385/1 and in A/174 top right). The printing racks drive the printing wheels with which they mesh: each of the thirty racks is lowered by a distance proportional to the associated figure wheel value. Lowering the rack drives the associated print wheel to register the relevant digit value. The thirty printing wheels align to give a line of type from which an inked impression is taken during the print cycle.

The printing wheels mesh with horizontal racks that transfer the digit values of the result to two rows of thirty punch wheels, one small, one large (Fig. 4.3). At the appropriate point in the cycle the punch wheels detach from the horizontal racks and are lowered to impress their settings into the soft material held in trays below to provide a stereotype mould.

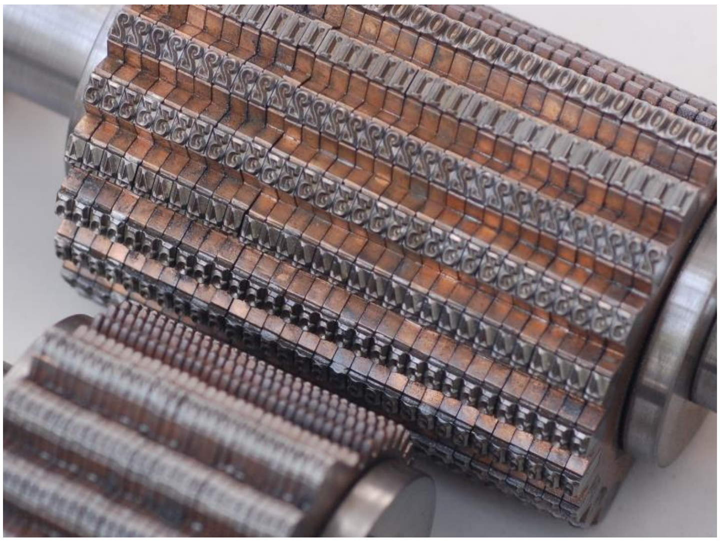


Fig. 4.3: Stereotyping punch wheels, large and small.

A full 30-digit result is impressed in one action i.e. all digit positions are impressed simultaneously. (Babbage calls the number wheels 'stereotyping sectors' (A/172) as only a portion of the circumference of each wheel has embedded in it ten heads embossed with the numerals 0-9. In this account the number wheels are called punch wheels. The punch wheels have two degrees of freedom only: rotational and vertical: each punch wheel can rotate about a horizontal axis to position the correct digit, and they separate from the horizontal racks when they lower to impress the result. The punch wheels do not move in the horizontal plane. Instead the stereotyping trays are repositioned each cycle to receive a new result in a fresh location. The apparatus provides two type-sizes, large and small. The two print pitches are fixed and are unalterable at 1/8" and 1/16". The two sets of print heads act in tandem: both are lowered together regardless of whether one, the other, or both print-heads actively impress. So, large and small impressions can be taken in the same action, or one or another set of punch wheels can be used separately.

The overall action of the transmission train between the calculating section and the output apparatus is to transfers circular motion of the figure wheels, which rotate in a horizontal plane, to circular motion of the printing wheels and to two sets of punch wheels, all of which rotate in a vertical plane.

Main Drawings: A/172, A/173,
Related Drawings: A/147, A/161, A/162, A/163, A/164, A/165, A/166, A/174, A/175, A/176

4.1 Transferring Results from the Calculating Section

At the start of the second half-cycle the tabular figure wheels (even axis) transfer digit values to the last sector axis during evens-to-odds addition. Each sector rotates anticlockwise (in plan) by the amount of the figure wheel digit value as the figure wheel is reduced to zero. Instead of giving-off to the next odds figure wheel axis as do all the other even axes, here the tabular axis gives off to the compound racks via the sector wheels with which they mesh (Figs. 4.2, 4.4, 4.5).

Each of the compound racks has two sets of teeth at right angles: vertical teeth in the horizontal plane that mesh with a sector wheel, and horizontal teeth in a horizontal plane that mesh with an associated pinion (elevation A/174, Figs. 4.5, 4.6). The pinions drive a set of horizontal spindles fitted with another set of pinions at the far end. These mesh with a set of vertical racks (printing racks) shown in elevation in A/174 (left) and in plan in A/173 (centre) (Fig. 4.7). The overall function of the arrangement is to transfer the circular displacement of the tabular figure wheels into proportionate vertical displacement of the racks.

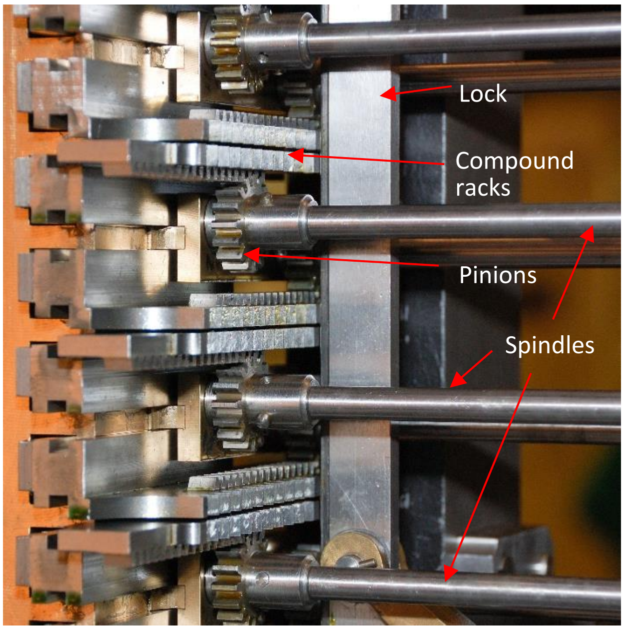


Fig. 4.4: Compound racks, pinions, spindles and vertical Lock. View from rear (detail).

The spindles are arranged vertically in pairs in two runs (Figs. 4.5, 4.7). The spindles for odd-numbered digits, 1, 3, 5, 7 . . . corresponding to units, hundreds, ten-thousands . . . form the run closest to the front of the Engine and engage with the racks below ($^{2}\boldsymbol{N}$ A/172 top centre, A/174, and A/176); spindles for even-numbered digits, 2, 4, 6, 8 . . . . . corresponding to tens, thousands, hundred-thousands . . . . are towards the rear of the Engine and engage with the racks above ($^{3}\boldsymbol{M}$).

The printing racks for the odd-numbered digits are back-to-back with those for the even-numbered digits (A/172 elevation top centre, A/173 plan, centre). Each of the pinions engages with a short run of teeth on the corresponding printing rack and both the pinions and the rack teeth are vertically staggered as shown (A/174 left, Fig. 4.7).

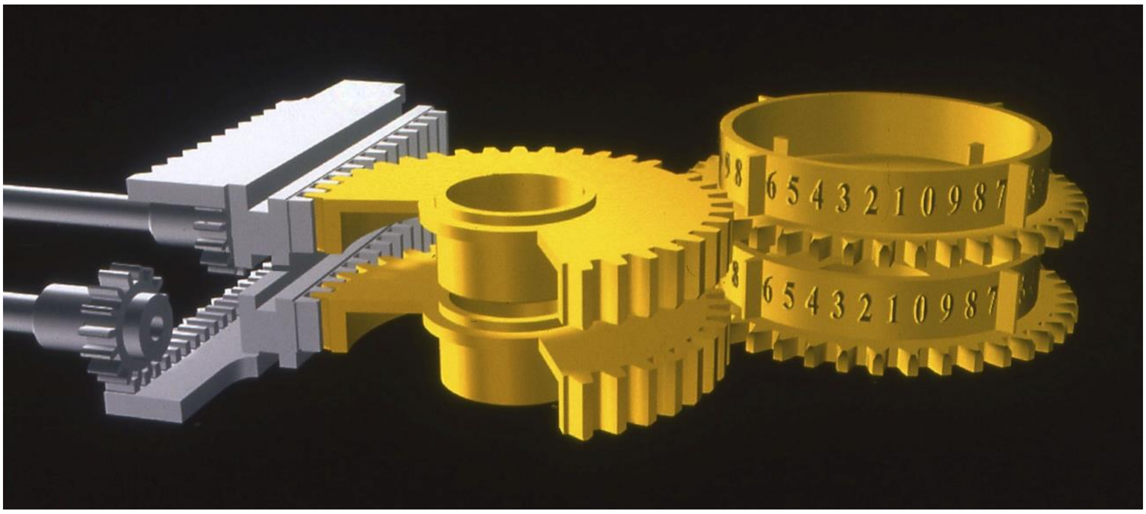


Fig. 4.5: Transfer of tabular values from figure wheels to horizontal spindles (simulation).

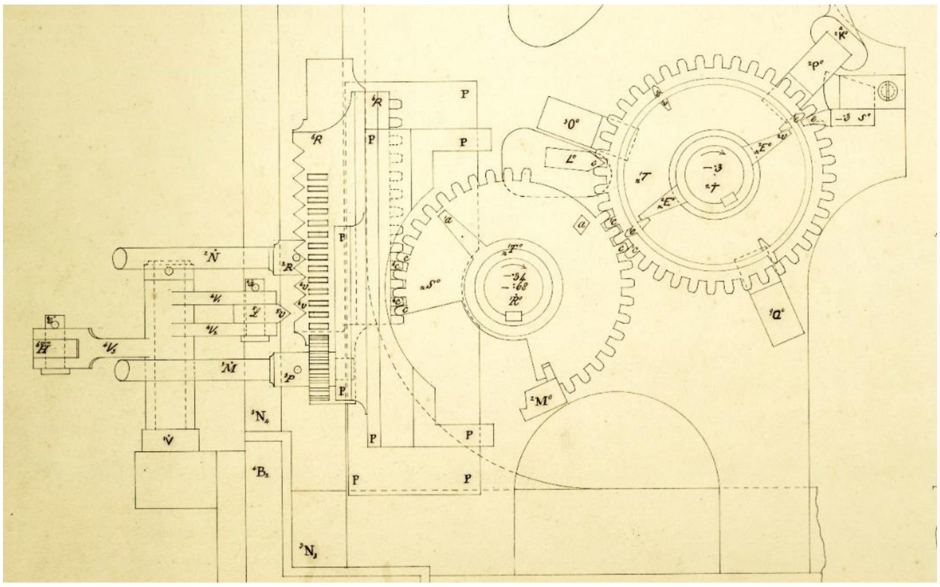


Fig. 4.6: Transfer of tabular values to horizontal spindles (A/176) (detail).

The drive train in elevation in A/174 shows the four lower-most tabular figure wheels (units, tens, hundreds and thousands) coupled digit-for-digit to the printing racks via the transmission train. The upper section of the tabular column is not shown. While the tabular column records a thirty-one-digit result there are only thirty printing racks shown in A/173 and elevation A/174 i.e. there is no provision in the drive train for the top-most digit and this value is neither transferred nor printed.

The output sectors ${}_n\boldsymbol{S}^0$ (A/176, Fig. 4.5) act as normal even sectors. Their drive and control are identical to that of the even sector axes for the 2nd, 4th, and 6th difference sectors i.e. they give off (to the transmission train) during addition of even to odd differences, restore the tabular value to the tabular figure wheels by reducing to zero, and at the same time return the compound racks and printing racks to their home positions. However, the output sectors differ from those of the other three even axes sectors ($\boldsymbol{S}^2$, $\boldsymbol{S}^4$, $\boldsymbol{S}^6$, A/176) in respect of depth of teeth, number of teeth, and angular orientation.

The opened-out view at the top of A/171 (Fig. 3.3) 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 those on the left are single depth, to allow partial disengagement in the intermediate vertical position (see **3. Calculation**, **Sector Wheels**, p. 27). By way of difference both runs of teeth on the output sectors (${}_n\boldsymbol{S}^0$) are full depth (Elevation A/174 top right). Here the sectors, shown fully raised, are disengaged from the tabular figure wheel but still engaged with the compound racks. Unlike the other three sector columns the ${}_n\boldsymbol{S}^0$ sectors remain fully engaged with the racks throughout, whether or not engaged with the figure wheels. The need to fully disengage the sectors from the lower order difference columns was to allow successive additions to accumulate with each successive cycle i.e. disengagement allowed the sector to restore the higher order difference without disturbing the lower order difference that had just been the beneficiary of giving-off. In the case of the output sectors (${}_n\boldsymbol{S}^0$) there is no such need and the printing racks need to remain fully engaged (via the transmission train) with the output sectors to return the racks to the home position and to track sector rotation as the sectors register results.

A second difference is that the output sectors have one more tooth than the standard sectors. A/176 bottom left (Fig. 4.6) shows 13 sector teeth for engagement with the compound rack while there are 12 teeth on other axes for the same section of the sector.

Finally, the angular orientation of the sector teeth that engage with the compound rack is different (A/176, A/161): A/161 shows the limits of travel of the output sectors as 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 on the output sectors (${}^2\boldsymbol{M}$) is shown scalloped out to accommodate maximum displacement of the sector (A/161, bottom left). The amount of rotational travel of the output sectors is the same as that for the difference column sectors. Only the orientation differs.

Locking the Compound Racks

The compound racks driven by the figure wheel sectors are shown in plan in A/176 (bottom left, Fig. 4.5) and in elevation in A/174 (top right). These views show a single wedge-shaped lock ($^{5}\mathscr{L}$ A174 'Locking Bar'; also A/176) operating on the vertical stack of thirty racks. The operation of the lock is driven by two of the fourteen cams in the output apparatus cam stack. The lock is lifted and withdrawn by the angle crank ($^{4}\boldsymbol{V}_{3}$, $^{4}\boldsymbol{V}_{2}$, A/174) turning on fixed pivot $^{1}\boldsymbol{V}$. The crank is driven by link $^{6}\boldsymbol{H}$ (A/174) which is in turn driven by the cam follower rocker arm from conjugate cams $^{2}\boldsymbol{A}_{2}$, $^{2}\boldsymbol{A}_{3}$ (A/173). The cam profiles and follower arms ($^{1}\mathscr{V}_{2}$, $^{1}\mathscr{V}_{3}$) are shown in A/172 as part of the general arrangement, and abstracted more clearly in A/175. The axis of the pivots at each end of link $^{6}\boldsymbol{H}$ are at right angles to each other – a questionable practice but the assumption is that the deviation from linear drive is sufficiently small to be accommodated by play in the pivot.

The lock operates once each calculating cycle engaging at around 252° (Timing Diagram 337 X 21) and releasing at about 290°. Inked and stereotyped impressions are taken during this interval i.e. while the compound racks are locked to freeze the current result.

A/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 levers ($^{4}\boldsymbol{V}_{1}$, $^{4}\boldsymbol{V}_{2}$ A/176) and mounting bracket (A/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.

4.2 Printing Racks

The compound racks drive the next part of the transmission train i.e. the spindles. Pinons at Engine end of the spindles mesh with the compound racks, and at the far end the mesh with the printing racks which are displaced vertically to transfer the figure wheel values to the printing wheels and to the stereotyping punches (Fig. 4.7).

Each printing rack has two short runs of teeth, one set in the upper section, the other in the lower section. The teeth in the upper section mesh with the pinions pinned to the horizontal spindles from the calculating section (A/174); the lower teeth mesh with the 'printing sectors' or printing wheels (A/172 centre, Fig. 4.9). The overall arrangement transfers the circular motion of the tabular figure wheels to circular motion of the printing wheels.

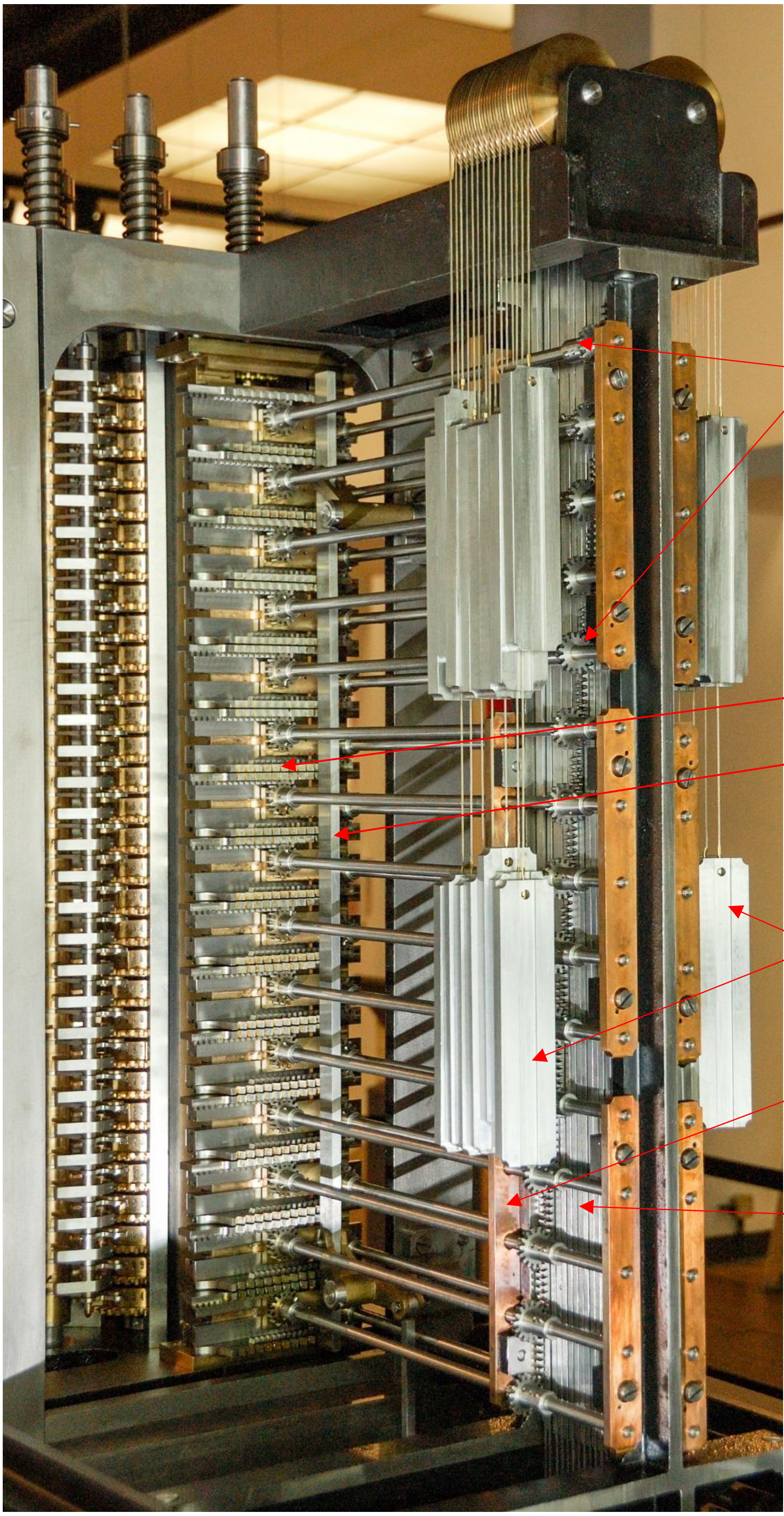


Fig. 4.7: Compound racks, pinions, spindles and vertical printing racks. Oblique view, rear.

Only the lower section of the racks is shown (A/174) and it is assumed that all thirty printing 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** (A/174) which act as constraining cheeks. The racks are further constrained by a guide-bar (**U**) which threads through slots at the lower end of all the racks. The guide-bar acts as a tie-rod for the assembly and constrains the vertical motion of the printing racks (A/174). The guide-slots in the racks and rectangular cross-section of the guide bar is shown in A/172 centre.

The nominal width of the rack pinions is 1/4" and the pitch of the racks is 1/8" (eight racks per inch) i.e. the pinion teeth are nominally double the width of the of the rack teeth. The increased width for the rack teeth is achieved by lapping adjacent racks in two mating L-shapes. Detail of the lapping is shown in A/173 (centre) (Fig. 4.8) which shows two pinions (digits one and two) and three lapped pairs (digits one through six) in cross section. The

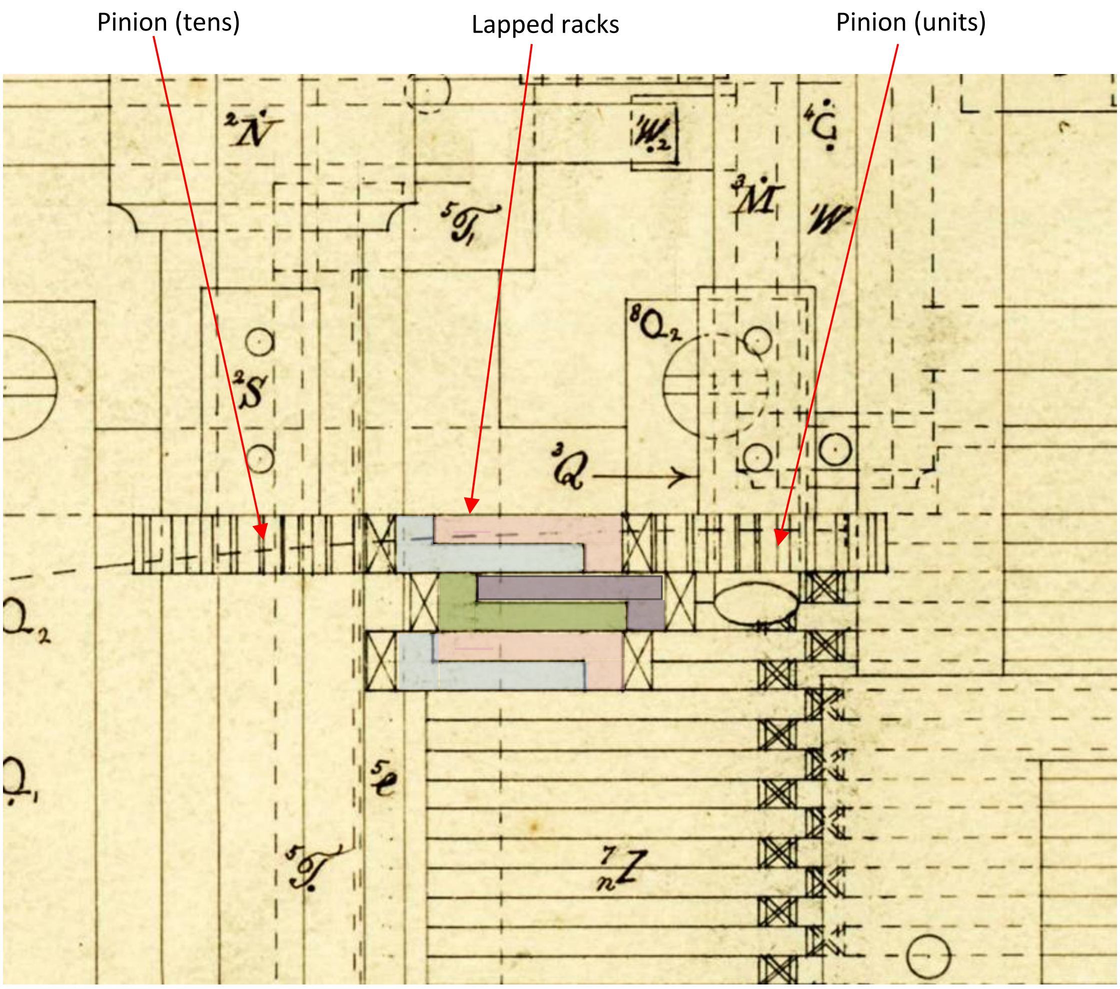


Fig: 4.8: Lapped racks (A/173) (detail) (colour added).

effect of lapping is to double the effective width of the gear teeth to twice the rack width. With lapping, the rack teeth and pinions for the fifteen odd-numbered digits face the front of the engine, and those for the fifteen even-numbered digits face the rear. Because odd-digit spindles are driven by the compound racks from below, and even-digit spindles from above, the odd spindles rotate in opposite directions for the same rotational direction of odd and even figure wheels. With the printing racks back-to-back this is corrected so that the clockwise rotation of the tabular figure wheels when reduced to zero, results in a uniform downward displacement in the related printing racks for all digit positions.

In addition to lapping, alternate lapped rack pairs are shown offset (Fig. 4.8). 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 – this to reduce the risk of pinions fouling adjacent racks due to side-play in the spindles. The offset is shown as carried through to the printing wheels for which the racks act as guide channels.

The upper sections of the racks are geared only where needed as shown in A/174 i.e. each rack has a short run of teeth positioned to mesh with its drive pinion. The same holds true for the lower rack sections in A/172. The positions of the toothed sections on the upper portions of the racks are staggered diagonally bottom right to top left across the face of the rack assembly (A/174 left and Fig. 4.7). Printing racks are therefore not identical (337 K 332 A-Q).

The lapping arrangement was retained though offsetting alternate racks was ultimately dispensed with and each of the faces of the racks are flush i.e. in the same vertical plane (see **Implications of Rack Offset**, p. 61). With the racks no longer offset the staggering of the spindles was also dispensed with.

The upper section of the rack assembly is not shown in A/172. The guide bar assembly, which constrains the lower section of the racks, was not duplicated higher up. Rather the upper section of the racks is trapped front-to-back between the two sets of pinions which act as a restraining cage (A/172, top centre).

Length of Lapping

A/174 shows thirty racks 1/8" wide and pinions nominally 1/4" wide. In the L-shaped lapping discussed earlier the fifteen odd-numbered racks (digits 1, 3, 5, ... , 29) face the front (A/173 right. The least significant digit is represented on the right-most rack in A/174), and the fifteen even-numbered racks face the rear. A/174 has solid lines at 1/8" pitch to represent the rack members. This would indicate that only the rack sections that are toothed for meshing with pinons are lapped: if the lapping was extended beyond the toothed section the alternate vertical lines should 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. Lapping was extended beyond the toothed sections upward and downward but stopping short at the lower end where the printing wheel racks revert to a true 1/8" pitch i.e. with no lapping flange. Details of the racks as made are shown in 337 K 332 A–Q.

Implications of Rack Offset

The printing sectors (or printing wheels) are concentric and pivot on the same shaft (Fig. 4.9) so offsetting alternate racks alters the pitch circle of alternate sectors. Equal downward displacements of the printing racks will therefore result in different angular displacements of alternate printing wheels. 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 '0' and one set to print a '9'. If the pitch of the type heads was identical for all the printing wheels then the '9' would be positioned as much as a full character height above an adjacent zero. This difference is clearly too large to be ignored. To ensure that the type heads line up for equal rotations of odds and evens spindles would require altering the pitch of the type heads on alternate printing wheels.

The locking arrangement would also need to be altered. A V-shaped locking bar inserts into notches across the run of printing wheels (Fig. 4.9). To ensure that the common locking bar will align with the V-notches whatever numbers are registered on any of the printing wheels, the pitch of the locking notches would need to be altered for alternate printing wheels. Only one printing wheel is shown in elevation A/172 and there is no indication of a call for non-identical printing wheels to take account of the printing rack offset shown in A/173.

If the purpose of offsetting alternate printing racks was to create channels whose sidewalls would confine the pinions and prevent them fouling adjacent racks, then it could be argued that the same design-need (or precaution) should apply to the arrangement for meshing the printing wheels with the horizontal racks for transferring results to the two sets of stereotyping punches (A/172 bottom). The teeth of the printing wheels remain in engagement with the upper teeth of the horizontal rack throughout. However, both sets of stereotyping punch wheels separate from the lower teeth of the horizontal rack when lowered to make an impression in the tray below. Extending the benefits of the channel-guide effect of offsetting alternate racks would assist alignment when the punch wheels are raised to re-engage.

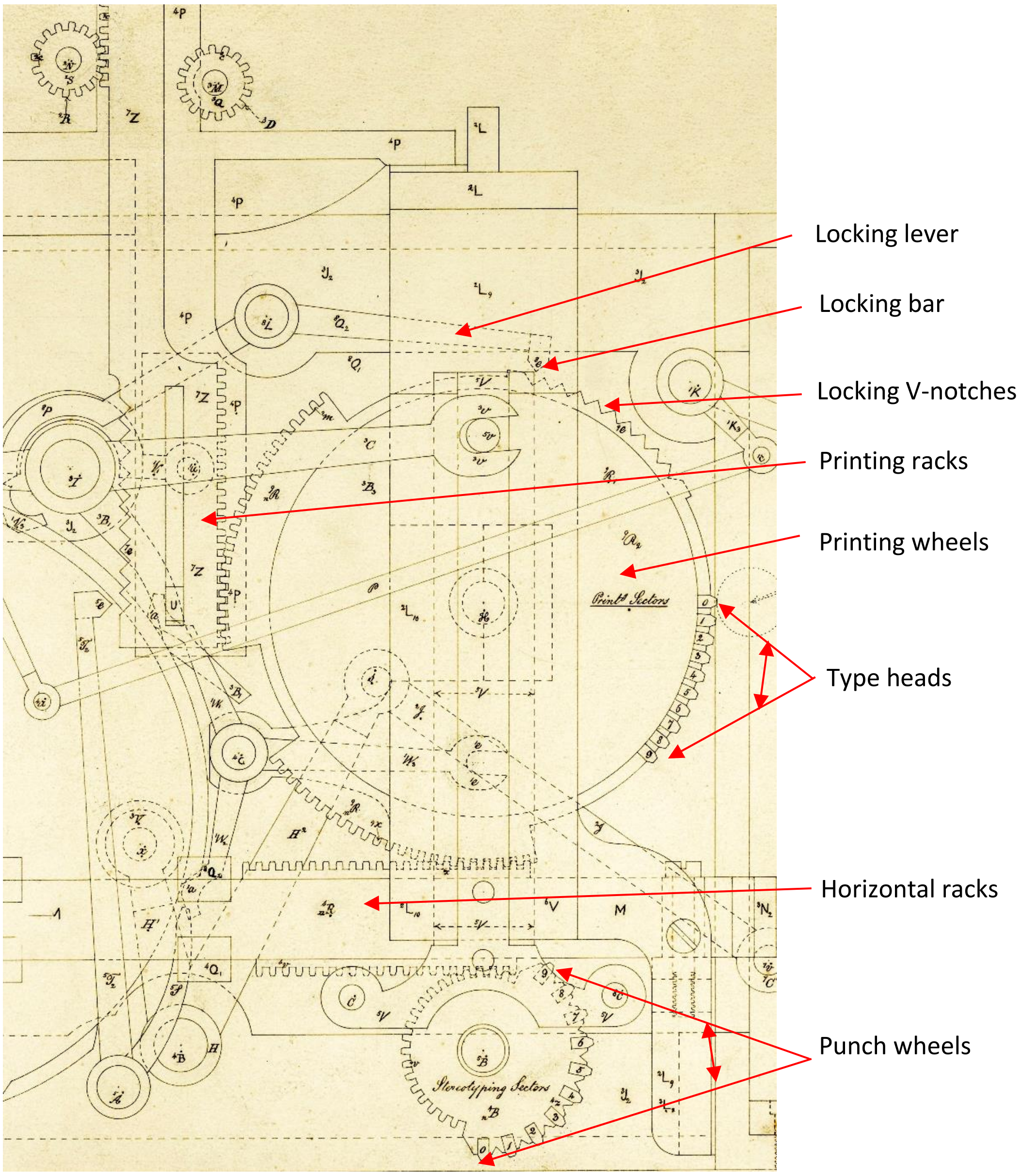


Fig: 4.9: Printing wheels and racks (A/172) (detail).

The implication of offsetting alternate horizontal racks is that the pitch of the type heads would need to be different for alternate punch wheels to ensure that they line up at six o'clock for all the sectors whatever the number set up. As with the printing wheels, there is no evidence in A/172, or elsewhere, of provision for non-identical stereotyping punch wheels. The implications of printing rack offset do not seem to have been carried through to the train for setting up the printing wheels or stereotyping punches, though the printing-rack offset is explicit in the drawings.

The issue of how far into the drive train to extend the offset is resolved by the geometry of the stereotyping punch wheels. In the case of the printing wheels the run of locking notches is separate from run of type heads and the pitch of each of these can be separately fixed. However, in the case of both small and large stereotyping punch wheels the locking notches alternate with the type heads between which they interpose ($_{n}{}^{4}\boldsymbol{B}$, ${}^{5}\boldsymbol{B}$, A/172). Increasing the pitch of the type heads on alternate punch wheels 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 bars of the crab-claw lock on the smaller set of punch wheels (twenty-past the hour in A/172) which operate across all the wheels i.e. the two separate conditions (the rotational position of the punch wheel, and a fixed separate locking point) cannot both be satisfied if the pitch circle of alternate sectors differs and the type heads alternate with the locking notches. (The crab claw lock shown for the small punch wheels on the left is duplicated for the punch wheels with larger type on the right (A/172) though this is not drawn.) The duplication is inferred from the provision of the dovetail slide and driving mechanism from the cams (see **5.4 Locks and Modifications to Printing Racks,** p. 93).

In the case of the printing wheels the two conditions are reconciled by placing the sequence of type heads and locking notches into separate groups. In the case of the stereotyping punch wheels there is insufficient space on the circumference of the significantly smaller stereotyping punch wheels to accommodate separate runs of type heads and locking notches.

The upshot is that the printing wheels and horizontal racks, if implemented as drawn, must be offset to correct for differing displacements of alternate printing wheels resulting from offsetting alternate printing racks; but the offset should not be extended to the meshing of the horizontal racks with the two sets of punch wheels on the underside. Alternate printing racks, printing wheels and horizontal racks are therefore not identical, while punch wheels for small type are identical as are the punch wheels for large type (337 K 383 A&B).

Dispensing with Rack Offset

The length of the pinion bushes is about 2.5". Four of these are shown in A/173 (top and middle, centre), two at the Engine end and two (the first two) meshed with the racks. Being the first two, their bosses are outside and clear of the rack assembly. For the undrawn pinions there is barely any clearance between the bushes of the recessed pinions and the adjacent pair of racks that form the cheeks of the guide. This is more clearly shown in A/172 centre top which shows minimal clearance between the cross section of the innermost spindle (^3M) and the teeth of the adjacent rack. Thinning the wall of the bush to 1/16" would provide workable clearance but such a reduction was not thought prudent.

The simplifying solution adopted was to abandon offsetting alternate printing racks. Slightly reducing the width of the printing racks' pinions and chamfering the teeth were considered sufficient precaution against pinions fouling adjacent racks.

The lower rack sections, which mesh with the printing wheels, do not have the benefit of lapping and the rack teeth are only 1/8" across. Here the offset feature was retained to provide sidewall guidance for the printing wheels. The benefit of offsetting is to reduce, by providing guide channels, the need for critical manufacture and fitting. In the case of the pinions, sufficient margins of clearance are available to justify removing the offset feature and this solves the issue of wall-thickness of the pinion bushes. In the case of the printing wheels, the offset was retained and the lower sections of racks are not identical for even and odd powers of the digit positions (337 K 331 A-Q, 337 K 332 A-Q).

Counterbalancing the Printing Racks

There are two points in the cycle where there is a risk of runaway i.e. of uncontrolled downward motion where the racks lower under their own weight. The upper sections of the printing racks are not shown in A/174 but all thirty printing racks are assumed to be the same length and run the full height of the engine. Each rack is in light contact with its neighbours over their length and coupling through sliding friction was foreseen. The contact surface between racks includes the additional surface from lapping adjacent pairs (Fig. 4.8). The racks are made of steel and their collective weight in their unmodified form is 41.4 lbs.

The printing wheels are returned to zero at the end of each cycle (Timing Diagram 337 X 21). In this position all printing racks are fully raised – the position shown in A/172. The final (even) sector wheel is in its rest position i.e. full anti-clockwise (A/176) in its fully raised position. Lifting the sector into the fully raised position lifts them clear of the figure wheels and at the same time engages the sector lock (see **3. Calculation**, **Operation**, p. 32). With the sectors locked the printing racks are supported by the sectors via the transmission train made up by the sector wheels, compound racks, spindles and pinions (Fig. 4.2). The printing racks remain so supported until the start of the second half cycle i.e. the adding of even differences to odds.

The first point at which there is a risk of runaway is during the transfer from the last even sector column to the compound 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 compound racks (Fig. 4.6, A/176 detail) during the second half-cycle i.e. during even to odd addition. The second half-cycle starts with the tabular figure wheels, in general, displaced anticlockwise from zero), even sectors fully raised (locked and disengaged from the figure wheels), and figure wheels locked.

The even sectors then fully lower to engage the figure wheels, figure wheel locks withdraw, and figure wheel axes and zero stops lower. The expected next action is that the internal arms of the figure wheel axis will drive the figure wheels to zero and, in doing so, drive the printing racks via the sector wheels and transmission train in a controlled way. However, when the figure wheel locks withdraw there is a risk that the racks drop under their own weight and run ahead of the figure wheel internal drive arms. The effect would be for the falling printing racks to drive the transmission train (compound racks, sector wheels, and figure wheels) until the outer nibs of the figure wheels hit 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. For figure wheels in positions other than 9 the distance the transmission train will be driven will be in proportion to the figure wheel value. There is no evidence in the original design drawings for measures taken to prevent runaway.

The printing racks are locked only in the timing window in which an inked impression is taken and are unlocked for the rest of the calculating cycle (Fig. 3.2, 337 X 21). The locking period is later than the short window of vulnerability and so the rack locks offer no protection against runaway.

For the less mobile racks, friction-coupling would act to brake runaway neighbours. Equally, friction coupling could produce an avalanche effect as less mobile racks join the descent.

Runaway does not introduce either calculation or printing errors. The effect is simply to accelerate the transfer of result values to the printing racks in an uncontrolled action ahead of the rotation of the drive arms of the figure wheels, and for the figure wheels to impact the zero stops with more than necessary force.

The second risk of runaway occurs during restoration after stereotyping. Once the tabular value is transferred to the printing/stereotyping wheels, the printing-rack locks and figure wheel locks secure the train and immobilise the mechanism while impressions are taken (see Fig. 3.2 for calculation timing; F/385/1 for both calculation and output apparatus timing). The locks then disengage, and the intended next action is for the sweeping sector arm to engage with the drive pegs on the upper side of the sector wheels to restore the tabular value and return the even sectors and printing wheels to zero. This is the second risk-point for runaway. In general, individual sectors and printing racks will be displaced by different amounts from zero during the course of a calculation. 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 introduces calculating or printing errors. In both cases the restoring action of the drive arms will complete correctly. However, ramming of the mechanism against the zero stops in the first case, and against the rotating sector restoring arms in the second, is undesirable. There was also concern about the load on the sector restoring arms of the racks when lifted after printing, as well as the overall load on the main manual crank when the racks are collectively lifted and restored to zero.

To avoid the risk of runaway the precaution was taken, pre-build, to counterbalance the deadweight of the racks by providing individual counter-weights for each rack. There is no evidence in the original design for measures to prevent runaway.

Pulley-suspension for counterbalancing weights

Contemporary practice favoured using individual weights suspended by cords wound on drums in the pinion shafts. Two factors weighed against this: the lowermost pinion shafts have insufficient clearance for the full downward travel of the weights. Also, the counterbalancing load is transmitted through the friction of the spindle bearings and pinion gearing, and this force is variable. The arrangement preferred uses individual counter-weights attached directly to each rack by a cord suspended over a pulley on a shaft above (Fig. 4.7). The weights are long steel strips (8" x 1.5" x 0.5") and weigh 1.38 lbs each. Each of two pulley shafts run left to right (front and rear) and carry fifteen 3"diameter phosphor bronze pulleys 1/4" wide (337 K 322). 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 this discourages the tendency to spin. The weights are suspended in two tiers i.e. fifteen weights in two tiers at the front (eight above and seven below) and fifteen in two tiers at the rear. Channels cut down the faces in contact (337 K 22 and K 21) allow paths for the cords to the lower tiers. The tops 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 (337 K 331, K 332).

Calculation confirmed that if the cords slackened momentarily when the racks were raised the cords might jump the pulley wheels. A nine-digit upward movement of the rack restoration corresponds to a lift of 2.827" in 60° of the cycle. In the first 0.01 seconds 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 printing-rack locks, compound racks, and figure wheel locks. Without counterbalancing, the locks would be inserted against the weight of the racks and any correction for small derangements would be made against this resistance. This additional load would accelerate wear as well as increase the shock-load on the cam. Additional wear and load were considered to be undesirable given the steep pressure angle of the lock cam profile and the inability (as was found) to turn the engine without counterbalancing of the locks with additional springs (see **7. Drive, Counterbalancing Axes and Locks**, p. 181). The provision of counterbalances for the racks was taken to be consistent with the practice, in the original drawings, of providing spring counterbalances for the figure wheel shafts.

Counterbalancing the Printing Racks – Re-evaluation

Before the build the main rationale behind the addition of counterbalancing weights was to prevent runaway i.e. uncontrolled downward motion of the racks as they free-fall under their own deadweight. It was also thought that the counterweights would relieve the load on the locks that operate on the figure wheels, the printing racks, compound racks, and the stereotyping punch wheel locks. The thinking here was that for small downward derangements the locks would be inserted against the weight of the racks. This would increase the load on the cams driving the locks and increase wear.
When the printing apparatus was assembled, fears about runaway proved ill-founded. Stiffness in the drive train proved to be the most serious difficulty of the build: driving the racks against resistance was the most enduring of the difficulties and, as it turned out, masked several other issues right to the last. Before remedial measures were taken, the forces required to drive the transmission train were sufficient to deform the internal drive pegs of the last sectors.

The main cause of stiffness was from sliding friction between the racks. Surface tension using any wet lubricant produced unexpectedly high resistance to sliding. Various wet and dry lubricants were tried without success. Running-in the racks using an electric motor that continuously exercised the racks was tried, also without success.

The solution adopted was a combination of skimming down the overall width of each rack, selectively reducing contact surface area by milling recesses in the faces of the racks and using dry graphite lubrication. Milling the racks to reduce thickness and provide relief recesses released internal stresses and the racks distorted badly, twisting in several planes. The racks were straightened by hand manipulating them to restore straightness.

With these relieving measures there was still residual stiffness that made runaway less of a hazard. However, the counterbalancing weights were still justified on at least three counts:

1. as a precaution against runaway as the racks wear in and become freer.
2. to relieve the drive load on the sectors to lift the racks to zero.
3. to ease the insertion of the locks.

The worst-case deadweight loading occurs with an all-nines result. In this case the sectors lift the full weight of all thirty printing racks for the full travel during restoration to zero. In the all-nines case there is no sliding between the racks i.e. they all move as a block. If there is a mix of digits, as is usual, the deadweight load is increased by the sliding friction between the racks. Here the load on any individual sector accumulates as the restoration proceeds, sweeping up the digits in descending order (i.e. 9s first) gathering racks which ramps up the load as zeroising progresses. If the individual drive trains for each or the digit values was entirely separate, as they would be if frictionless, then the worst-case load on each sector would be no more than the deadweight load of one rack. It is the unavoidable coupling of the racks to each other through sliding friction that increases the load on individual sectors beyond the load of a single rack. The worst-case of rack-to-rack frictional loading occurs when no two adjacent numbers are the same. Excessive loads would also cause ‘wind-up’ in the spindles i.e. elastic twisting that could produce timing lags in the transmission. Looking beyond the sectors, the contribution of lifting the racks, to the overall load on the main crank, is also a consideration. Getting the printing racks back to zero and the inserting the locks produced the largest stresses to the drive.

Counterbalancing the printing racks relieves the drive stresses and also relieves resistance to insertion of the locks if these are slightly deranged from their aligned positions. Ragged alignment of the printing wheels, despite the action of the counterweights, required remedial modifications to the printing-racks lock and prompted an extended understanding of its possible intended function. For rationale and modifications (see **5.4 Modifications to Printing Racks Lock**, p. 91).

4.3 Modifications to Spindle Assembly

The original design shows the spindles having a horizontal stagger i.e. alternate spindle are offset front to back. A/172 (top centre) shows four spindles, units and hundreds on the right ($^{3}\boldsymbol{M}$), tens and thousands on the left ($^{2}\boldsymbol{N}$). (The spindles for hundreds and thousands at the upper edge of the drawing are shown without pinions). The two spindles on the right are shown offset from each other, as are the two on the left i.e. the spindles in each of the two sets, one to the front and one to the rear, are not in the same vertical plane.

Staggering the spindles is a result of carrying back the offset of alternate printing racks ($^{7}\boldsymbol{Z}$ A/172, Fig. 4.9). Dispensing with the rack offset dispenses with the need to stagger the spindles which can now be in the same vertical plane. To reduce the risk of pinions catching on adjacent printing racks, pinion teeth were thinned to 3/16" to engage with a 7/32" wide rack-tooth. The compromise distance between the vertical centre-lines through the spindle centres is 2.160" (337 K 321 A & B, 337 K 311).

Removing the spindle stagger removes the need for the D-shaped scalloped cut-outs in the vertical framing pieces to accommodate the first few of the lower most pinion bushes – until the stagger takes bushes clear of the framing member. Examples of scalloping the frame are shown for two of the four pinions in A/172 (top centre).

There is also a correction to be made to the compound racks at the sector-wheel end as a result of removing the spindle stagger. The mirroring solution to the layout flaw (**3.5 Resolution of the Layout Design Error**, p. 47) in the basic addition mechanism required a 2¼° rotational correction applied to the sector and figure wheels so that the centre-line symmetrically bisected either a tooth or a tooth-gap for both odd and even axes. This is carried through to the smaller of the compound racks: the centre-line of the rack tooth gap is moved to the centre-line of the sector wheel i.e. not as shown in A/176. The corrected layout is shown in 337 K 312, 337 K 313.

Additional Spindle Bearings

The original designs show the spindles supported by bearings at each end: one set near the compound racks at the figure wheel end of the Engine fixed to the framing pieces (**P**, A/174 top right), and one set (**R**) on the far left of the framing for the printing racks (A/174 far left). Only the lower few of the thirty spindles are shown in A/174 and It is also not clear how the bearing strips (**R**) fix to the main frame (^{5}**P**) nor whether the strips are a single unbroken run or divided into sections, each for a group of spindles.

For ease of assembly and dismantling, six separate bearing strips were used, each drilled to take five spindles (Fig. 4.7). Three of these are mounted vertically in line on the outside of the vertical rack frame towards the front of the machine. These support the fifteen spindles for the odd power digits 10^1, 10^3 . . . , 10^{15}. The second set of three is located behind these and supports the spindles for the even powers (10^0, 10^2 . . . , 10^{30}).

The torsional load on the spindles depends on the stiffness of the train and the weight of (counterbalanced) printing racks and there was concern was that spindles might bow under excessive load. If the spindles bowed sufficiently, as they are likely to do should the racks jam or in the event of undue stiffness, there was concern that pinions might ride up over the rack teeth. It was thought preferable to allow the Engine to jam by preventing the spindles from bowing than for the pinons to damage the rack teeth by overriding.

As a precaution against bowing an additional set of six separate bronze bearing strips was fitted to better support the spindles (337 K 571 A&B). These mirror those already described and are mounted on the right side of the frame for the printing racks (Fig. 4.7). As described above the spindle stagger was dispensed with. The original and the additional spindle bearings are in the same plane i.e. the centres of the spindle shafts are vertically aligned.

If the spindle bearings were set in the cast frame the pinions would need to be unpinned to remove the printing racks. Dismountable bearings strips allow the spindles to be removed while keeping the pinning undisturbed. In the arrangement shown (Fig. 4.7) spindles can be removed in groups of five. The bearing strips are dowelled in place to for precise and repeatable registration.

5. Printing

The printing apparatus produces inked hardcopy on a paper print roll (Fig. 2.6, p. 16). The printed copy is used for checking results and as a record of the same results impressed in the stereotyping trays (Fig. 2.17, p. 16).

Each calculating cycle transfers thirty-digit results to the printing wheels from the figure wheels of the results column. Once the printing wheels are set up with a result there are three distinct operations that make up the print cycle:

1. Inking
2. Printing (taking an inked impression)
3. Paper feed.

The inking system is located immediately above the paper management mechanism (Figs. 5.1, 5.2). The inking process uniformly loads the surface of the inking roller with relatively high-viscosity ink held in an ink bath. Once loaded the inking roller arcs downwards to deposit ink on the line of type presented by the print wheels. After the inking roller has retracted to its

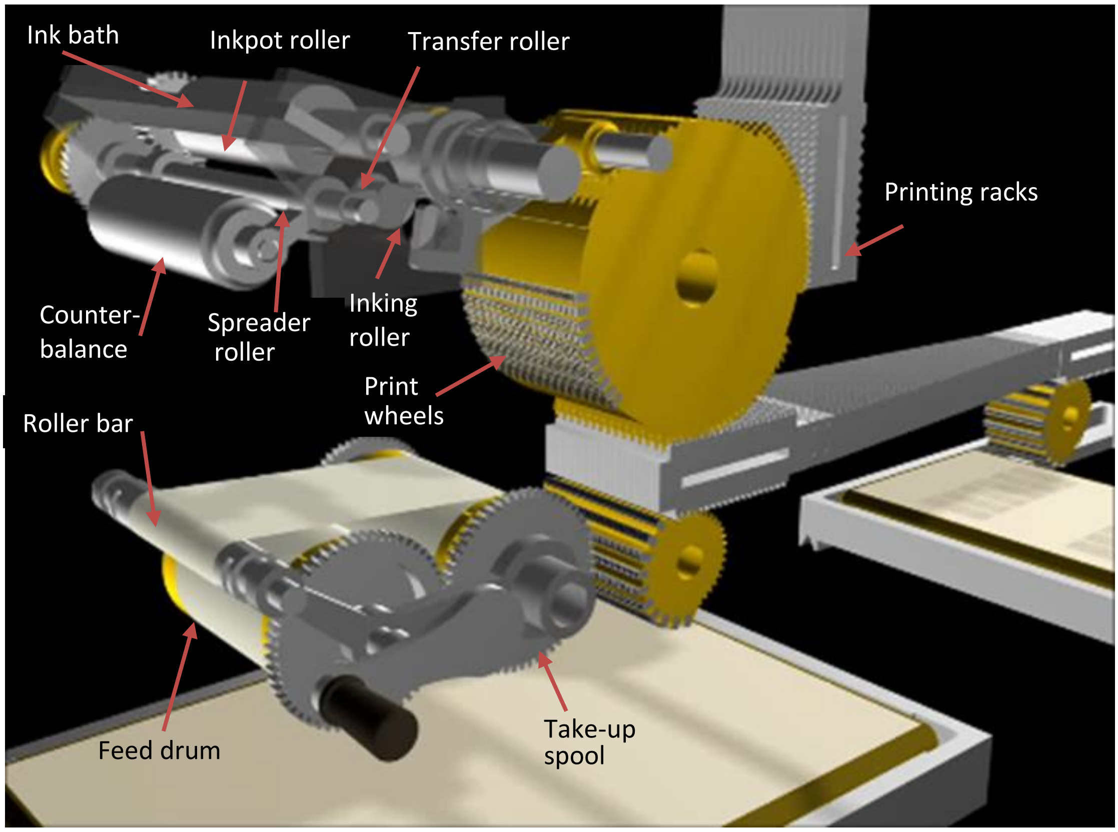


Fig. 5.1: Apparatus for inked printout (simulation).

raised position, the paper mechanism arcs upwards to take an impression by pressing the roller bar, around which the paper is looped, against the print wheels (Fig. 5.3). The paper mechanism then retracts to its lowered position. The retraction of the paper mechanism to its lowered position automatically advances the paper roll by one line in preparation for the next printing cycle. There is one printing cycle for each calculating cycle. There is no buffering or storage of results: each result is printed during the calculating cycle that generates it.

The layout of the printed results is fixed i.e. line-height, margins, and font are unalterable and results are printed one per line in a continuous single column with no blank lines between groups of lines (Fig. 2.6, p. 16).

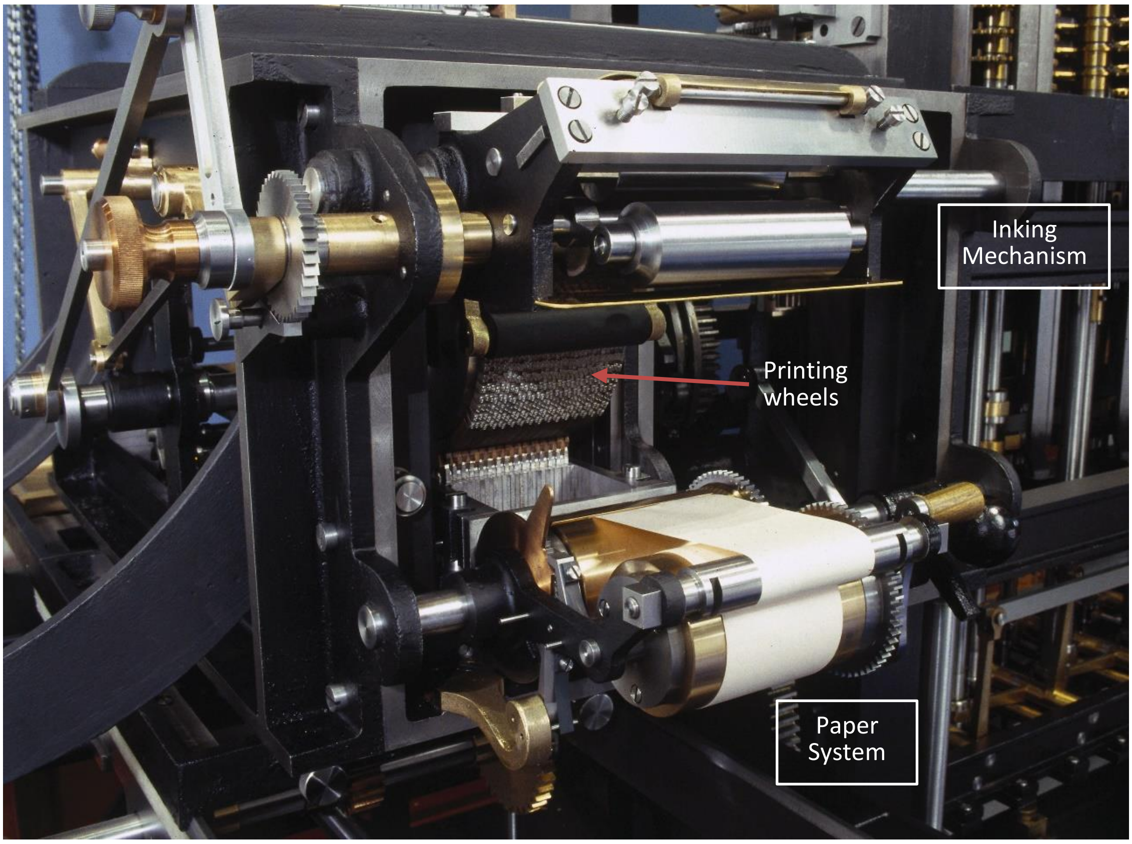


Fig. 5.2: Printing apparatus.

The general arrangement is illustrated in Figs. 5.1, 5.2. The inking, paper management, and printing mechanism is shown in the right-hand half of elevation A/172 (excerpted in Fig. 5.3 below), and bottom right plan A/173.

Main Drawings: A/172, A/173, A/165

Related Drawings: A/163, A/164, A/174**, A/**175, A/176.

5.1 Inking

The purpose of the mechanism is to ink the line of type presented by the printing wheels to which each thirty-digit result is transferred once each cycle.

The inking mechanism shown in Fig. 5.2 (A/172) incorporates a cluster of four rollers: the inkpot roller, transfer roller, spreader roller and finally the inking roller (Fig. 5.3). The ink supply is held in a reservoir formed by four components: the inkpot roller ($^{1}\boldsymbol{L}$**)** a scraper plate which bears on the axial surface of the roller to ensure uniform distribution of ink and to trap debris, and two side cheeks (not shown in Fig. 5.3). The ink is sufficiently viscous to prevent leakage past the scraper, an arrangement typical of its time. The ink bath has a flip-up dust cover with a curled lip for lifting.

The ink-pot roller bears on a second roller, the transfer roller ($^{5}\boldsymbol{J}$), which itself bears on a third roller, the spreader roller ($^{3}\boldsymbol{K}$), which makes the final transfer of ink to the inking roller ($^{3}\mathscr{J}$) (Fig. 5.3).

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 shaft $^{1}\boldsymbol{K}$ (A/172, A/173 top right) which is supported at each end by the two outer framing members ($^{1}\mathbf{J}_2$, $^{3}\mathbf{J}_2$) i.e. the shaft spans the full width of the apparatus. The arms and linkages driving the downwards sweep of the inking roller are shown in elevation A/171 top right. The shaft is driven by a lever arm ($^{1}\boldsymbol{K}_3$) itself driven by link $\mathscr{D}$ which pulls the roller onto the line of type. The arc of the roller trajectory is shown in A/172 (Fig. 5.3). (See **5.5 Cams, Drive and Control**, p. 96).

There is detail missing from the mechanisms as shown in the drawings. While the drive train for the inking roller is shown, there is no detail for the remaining three rollers with respect to drive, whether motion is intermittent or continuous, or the composition of the roller material. Missing detail was supplied before the build during the design phase, and other remedial modifications were introduced after the first commissioning trials as described below.

Modifications to Ink Feed

A review of mid-19^{th}-century printing practice indicated that inking would be too heavy if the feed was continuous. No provision is made in the drawings for a method of driving the inking system and an intermittent rather than continuous drive was provided for the inkpot roller to slow the ink feed.

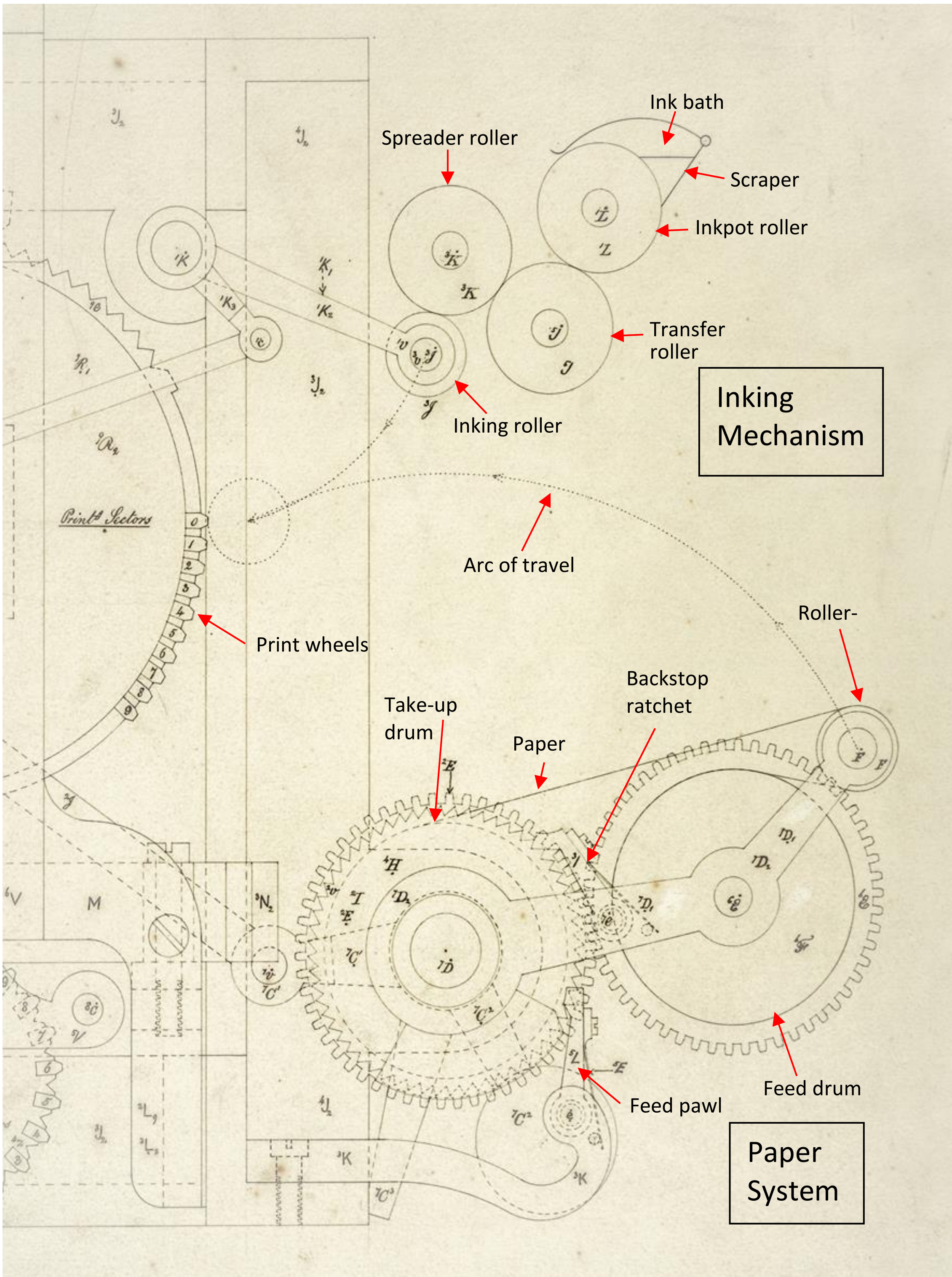


Fig. 5.3: Inking and paper system (A/172) (detail).

The drive for the inkpot roller is derived from an eccentric added to the main printer cam shaft ($^{2}\boldsymbol{D}$ A/172, 173 centre, Fig. 5.4) outside the inking roller drive arm $^{4}\mathscr{U}_{3}$ (bottom of A/173). The arrangement is shown in assembly drawing 337 K 25 top left. The reciprocating motion of the link operates a pawl which advances a ratchet (Fig. 5.5). 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 (337 K 446) using the knurled thumbscrew. This alters the pull-back of the pawl and the motion transmitted to the

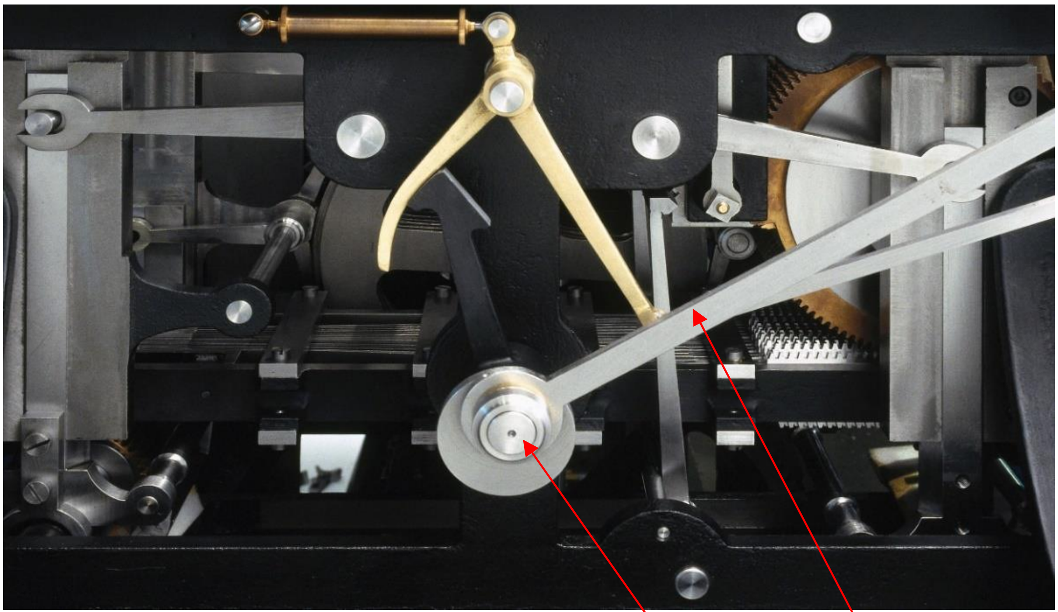


Fig. 5.4: Inkpot roller drive (before anti-bounce and printing racks lock modification).

Eccentric Reciprocating drive link

roller. The inkpot and spreader roller are actively driven from the ratchet wheel shaft via a short shaft and separate gear wheels i.e. the inkpot roller is not driven directly by the ratchet shaft but via gearing. The spreader roller is also geared to the ratchet drive shaft. Reverse rotation is prevented by a backstop pawl (337 K 448). The remaining two rollers are slaved (friction driven) i.e. do not have independent drives. The ratchet drive for the inking system was drawn up in the design phase i.e. in advance of the build.

It is not clear from A/172, A/173 or A/165 whether the roller bar ($\boldsymbol{F}$) is intended to rotate or not. A/172 suggests that it was intended to rotate driven by the paper advance. However, to reduce possible unevenness in ink density across the face of the type wheel from the curvature of the roller, the preference was to provide a fixed bar with a flat so as to act as an anvil for the type-heads. The flat consists of an insert strip made from composite material with some plasticity (Fig. 5.14).

Since the ink-feed ratchet drive is derived from the main output apparatus cam-shaft, itself geared to the main Engine drive shaft, the ink-feed runs all the time the printer is driven.

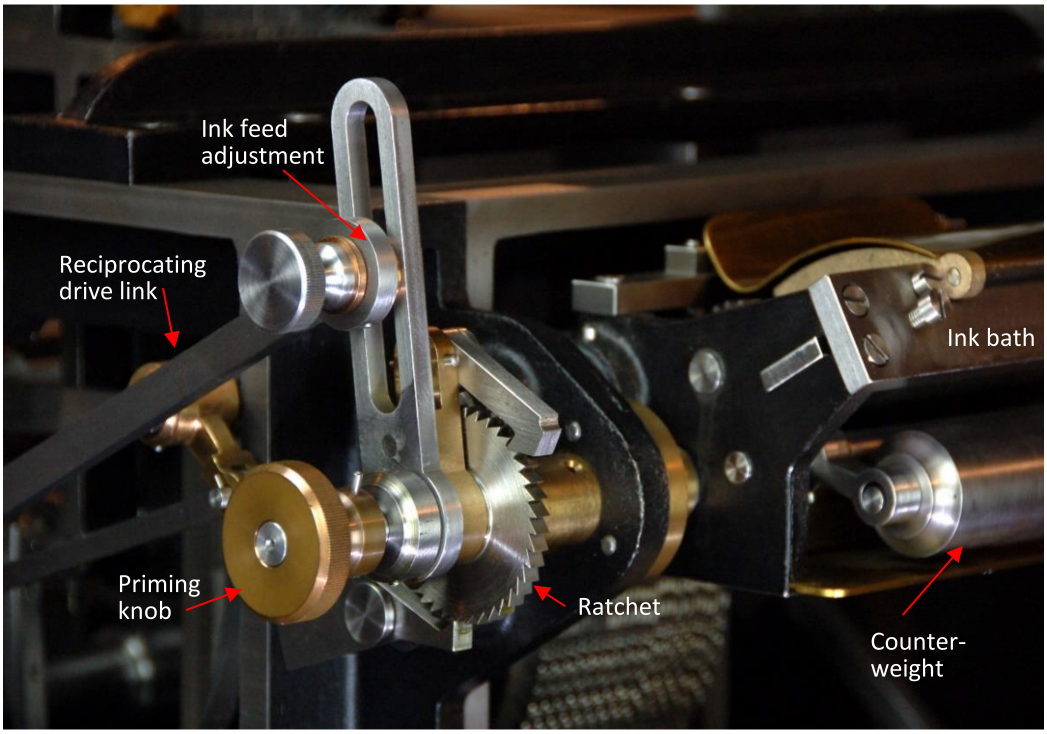


Fig. 5.5: Inking feed adjustment.

Spreader Roller

Contemporary 19th-century practice printing machinery evidences at least one roller in the inking train having reciprocating axial motion during rotation to ensure a 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 roller surface.

There is no provision in the drawings for a reciprocating sliding spreader roller, and an oscillating spreader mechanism was added at the design stage. This consists of two cams, one at each end of the roller shaft, that

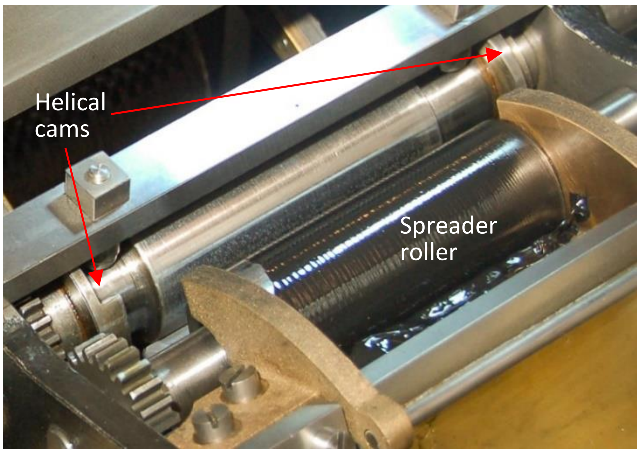


Fig. 5.6: Spreader roller reciprocating drive.

drive the shaft axially to and fro (Fig. 5.6. Assembly Drawing 337 K 25, parts 337 K 428, K 429). The cams are circular 360° cams with sections of left- and right-handed helices. The cams ride against two pegs fixed to a support bar above the roller. The angular position of the cams and position of the 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 spreader roller is gear-driven from a gear on the shaft of the intermittent motion ratchet drive for the inking roller i.e. it does not rely on rolling friction to rotate.

An additional contemporary practice was to alternate the consistency of the rollers: the final roller ($^{3}\mathscr{J}$) would be soft to ensure uniform inking of the print heads, the ink pot roller ($^{1}\boldsymbol{L}$) 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 possibility of eliminating the two intermediate rollers ($^{5}\boldsymbol{J}$, $^{3}\boldsymbol{K}$) and modifying the final inking roller ($^{3}\mathscr{J}$) to act as a spreader was rejected: there is no lateral room for cams between the swinging levers that support the inking roller, and the inking roller cannot be shortened as it needs to span the full width of the thirty print wheels (A/172). The need for a separate spreading roller requires an additional transfer roller to maintain the practice of alternating soft and hard rollers.

Since the final inking roller is soft, the spreader roller, $^{3}\boldsymbol{K}$**,** should be hard and the transfer roller, $^{5}\boldsymbol{J}$, soft. The soft roller has a less defined surface, is subject to greater wear, and relies on pressure to ensure even surface contact on the two rollers it couples. To accommodate these marginal variations from a true cylinder, the soft roller has a floating bearing (Fig. 5.7). The spreader roller, $^{3}\boldsymbol{K}$ is steel and the transfer roller, $^{5}\boldsymbol{J}$, is sheathed with a firm but yielding compound.

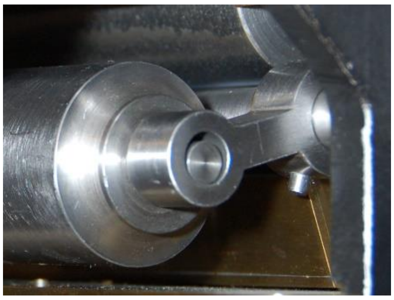


Fig. 5.7: Floating bearing.

The assembly of the inking system is shown in 337 K 25. 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 ($^{5}\boldsymbol{J}$) rests in a cradle which is part of a rocker mounting (detail 337 K 436). The transfer roller has a counterweight (Fig. 5.5) added (337 K 439) which allows the roller to bear with constant pressure on the spreader roller ($^{3}\boldsymbol{K}$) and inkpot roller ($^{1}\boldsymbol{L}$) . Raising the counterweight 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 (337 K 419) is mounted on bronze side cheeks (337 K 418 A&B) which act as the sidewalls of the ink reservoir, and also seal against the curve of the roller. The side cheeks insert into slots and the ink feed is adjusted by adjusting two jacking screws (337 K 423).

Anti-bounce Mechanism

The inking roller is driven in a downward arc (Fig. 5.3, A/172). The drive train consists of a rotating cam arm ($^2\boldsymbol{A}_{16}$, A/172, K575) keyed to the main printer cam shaft ($^2\boldsymbol{D}$) and which acts on the wish-bone contact follower-and-lever ($^4\mathscr{U}_2$, $^4\mathscr{U}_3$) connected to a long drive link ($\mathscr{D}$) the far end of which is connected to a short pivoting arm ($^1\boldsymbol{K}_3$) that pulls the inking roller downward onto the line of type. At the cam end the follower is driven against the action of compression spring ($^1\boldsymbol{F}$). The profiles of the follower and cam are such that the downward travel of the inking roller is relatively gradual while the return to the home position is a snap action as the rotating cam arm sweeps past the end of the follower and the compression spring is released.

The spring ($^1\boldsymbol{F}$ A/172) has two functions: it provides contact pressure between the inking roller and the sliding roller while in the home position, and it restores the inking roller to its raised position after the downward sweep to the type heads.

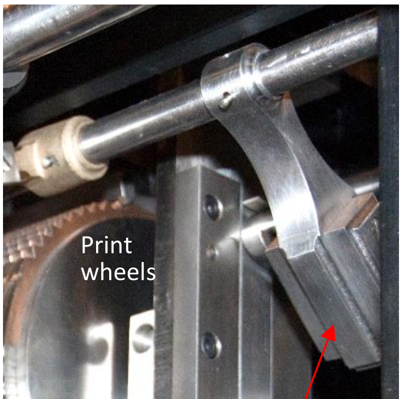


Fig. 5.8: Inking roller counterweight.

When retracting after the downward stroke, the inking roller has to clear the path of the paper roller on its upwards arc. The timing and the clearance are tight and during trials it was found that the inking roller was unable retract fast enough to clear the rising paper roller and the two fouled. To speed the return of the inking roller a counterweight (Fig. 5.8) was added to the shaft ($^1\boldsymbol{K}$) which drives the inking roller arms.The accelerated retraction allowed the inking roller to clear the print roller without fouling but, under the combined action of the spring and the counterweight, the inking roller rebounded off the spreader roller to again foul the paper roller i.e. the bounced inking roller caught behind the paper roller as the paper roller retracted and the two rollers jammed. An anti-bounce mechanism was devised and installed to trap the inking roller in its rest position against the spreader roller.

The principle of the mechanism is to trap the knuckle of the inking roller drive-link in its rest position to prevent bounce back (337 M 21 assembly). The main components are the release plate (337 M 396) and the bounce catcher (M 397). The anti-bounce mechanism is driven by the printer cam shaft, $^{2}\boldsymbol{D}$, (Fig. 5.4) which continuously rotates anti-clockwise while the Engine is

Fig. 5.9: Catcher plate (M397).

running (A/172). The catcher plate pivots freely on the cam shaft. The scalloped rear section (lower section in 337 M 397) acts as a counterweight and biases the catcher plate so that in its rest position the curved shoulder traps the drive link with the inking roller in its retracted position. The knuckle, in its trapped position, is on the curve of the catcher plate and stops the catcher plate from rotating anti-

Fig. 5.10: Release plate (M396).

clockwise any further. With the catcher in this position the drive link is inoperative i.e. it cannot retract to drive the inking roller on its downward arc. If not released, the next sweep of the cam arm would cause a jam.

The link is released by the action of the bronze release plate. The release plate pivots on a pin in the inkpot roller drive link (Figs. 5.10, 5.11) which, driven by the eccentric cam, reciprocates back and forth to operate the drive pawl for the inkpot roller (see **5.1 Inking, Modifications to Ink Feed**, p. 73). The motion of the drive link produces a compound motion of the release plate in which the plate moves anticlockwise and backwards to begin with the (the eccentric rotates anticlockwise) so as to hook the upper flat over the fixed release pin, and then clockwise and downwards to release the drive link from the shoulder of the catcher plate. The plate is released a little before the drive arm begins to propel the inking roller on its downward arc and remains in its released position until the knuckle of the drive arm retracts to begin the inking cycle. With the release plate displaced a few degrees clockwise, the knuckle of the drive link rides the curved shoulder of the catcher plate which is driven clockwise by the knuckle and held in contact with the

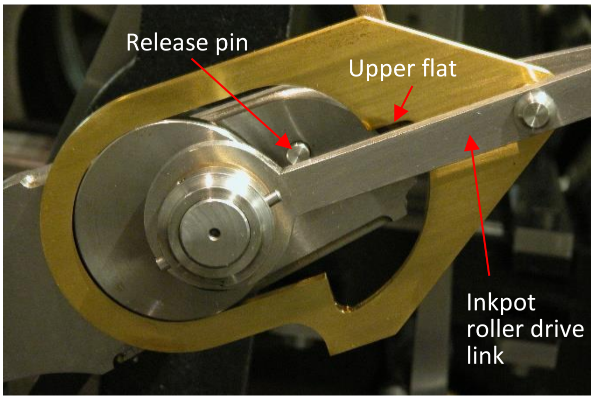


Fig. 5.11: Anti-bounce mechanism.

knuckle by the counterweight, which tries to rotate it anti-clockwise. The release plate moves forward so that the flat that depressed the release pin is well clear. When the inking roller drive-link, driven by the compression spring, snaps back, the counterweighted catcher plate drops back into the trapping position and the shoulder of the curve traps the knuckle in the rest position. The curved cutout between the two flats on the inside of the release plate is to give clearance for the release pin during the compound motion of the release plate.

Modification to Inking Roller Arms

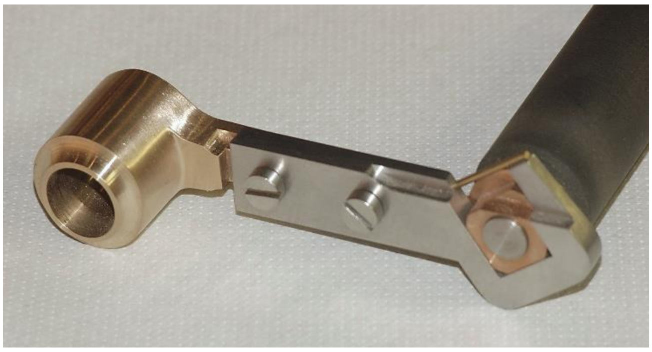


Fig. 5.12: Inking roller arms.

Trials showed that inking of the printing wheels was not uniform across the face of the print heads partly because the cylindrical inking roller was bearing on a plane surface, and ink deposition was denser along the centre-line of the printed result. The inking roller swinging arms were modified to allow adjustment of their length so that the line of impact along the type face can be adjusted up or down, and the ends of the arms were slotted to allow the roller to wipe upwards across the face of the print heads on contact pressure with the print heads (Fig. 5.12).

5.2 Printing Stroke

The general arrangement of the paper management system is shown in elevation A/172 excerpted in Fig. 5.3 and a partial plan in A/173. A/165 (bottom centre) has the most detailed views labelled *Fig. 7 Elevation* and *Fig. 6 End View*. *Fig. 7 Elevation* shows the mechanism in the fully raised position, as when taking an impression, as seen from the front of the Engine. *Fig. 6 End View* shows the mechanism as viewed looking left to right in solid line fully raised, and dotted in the rest position.

The paper path is shown in A/172 and A/165. Paper is threaded from the roll wound on feed drum $^6\mathscr{F}$, around the roller bar, $\boldsymbol{F}$, and onto the take-up drum $^4\boldsymbol{H}$ (Fig. 5.3). The relationship between the axes of the two drums and the roller bar is fixed i.e. the two arms, $^7\boldsymbol{D}_1$ and $^7\boldsymbol{D}_2$, carry the shafts of the drums and roller, and fix the relative positions. During the printing stroke the whole assembly moves as a unit i.e. there is no relative motion between the drums and the roller.

To print a result the whole mechanism turns on the fixed shaft ($\boldsymbol{D}^{12}$) and propels the printing roller in a circular arc 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 A/165, *Fig. 6*. The upstroke is driven from the single heart-shaped cam, $^2\boldsymbol{A}_4$ in A/175, and in plan as part of the cam stack towards the top of A/173 (see **5.5 Cams, Drive and Control, Printing Stroke Drive**, p. 97). A train of links and arms transmits the drive to the drum assembly. The angle of the elbow of the follower arm ($\boldsymbol{H}^1$, $\boldsymbol{H}^2$ A/172) is fixed and the two fixed pivots in the train are the follower arm pivot ($^4\boldsymbol{B}$) and the take-up drum axis, $^7\boldsymbol{D}$ (A/172, A/173). The two floating pivots are the forked knuckles at the ends of drive link $^2\mathscr{J}$. The follower arm, $\boldsymbol{H}^1$, pivots on $^4\boldsymbol{B}$ and drives link $^2\mathscr{J}$ diagonally downwards, and arm $^7\boldsymbol{C}^1$, keyed to shaft $^7\boldsymbol{D}$, levers the drum into the upstroke.

The assembly passes through and beyond top dead centre during the upstroke (A/165, *Fig. 6*) and the upstroke is halted by the roller bar meeting the print. The swinging arms with the two drums and roller-bar pass beyond the vertical as shown in *Fig. 6*, A/165. The drive is unidirectional i.e. the return stroke is not actively driven, and the assembly would remain at rest if the return stroke was unaided. To avoid the assembly being stranded in the printing position, a spherical counterweight is provided to initiate the return stroke. The counterweight, $^7\boldsymbol{C}^2$, is shown as a circle in elevation A/172 right bottom, and in plan in A/173 top right. The counterweight is fixed to a lever arm of one piece with the boss ($^7\boldsymbol{C}$, A/173). The counterweight 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 (A/175). During the return stroke the roller bar retraces the trajectory of the upstroke. The rest position at the end of the return stroke is determined by the end stop $^7\boldsymbol{C}^3$ which bears on the vertical frame (A/172 bottom right). Plan view (A/173) does not show the end stop lever but the notation indicates that it is part of the ball-weight lever assembly ($^7\boldsymbol{C}^2$, A/172).

5.3 Paper Feed

Of the feed drum and the take-up drum, the take-up drum is the more complex assembly consisting as it does of a series sleeves and drums rotating around each other. The main drawing for the paper feed system and the take-up drum assembly is A/165, *Fig. 7*. There are several details missing and aspects of the nested arrangement are not immediately evident. *Fig. 7* has been repeated here (Fig. 5.13) coloured to show the separate assemblies.

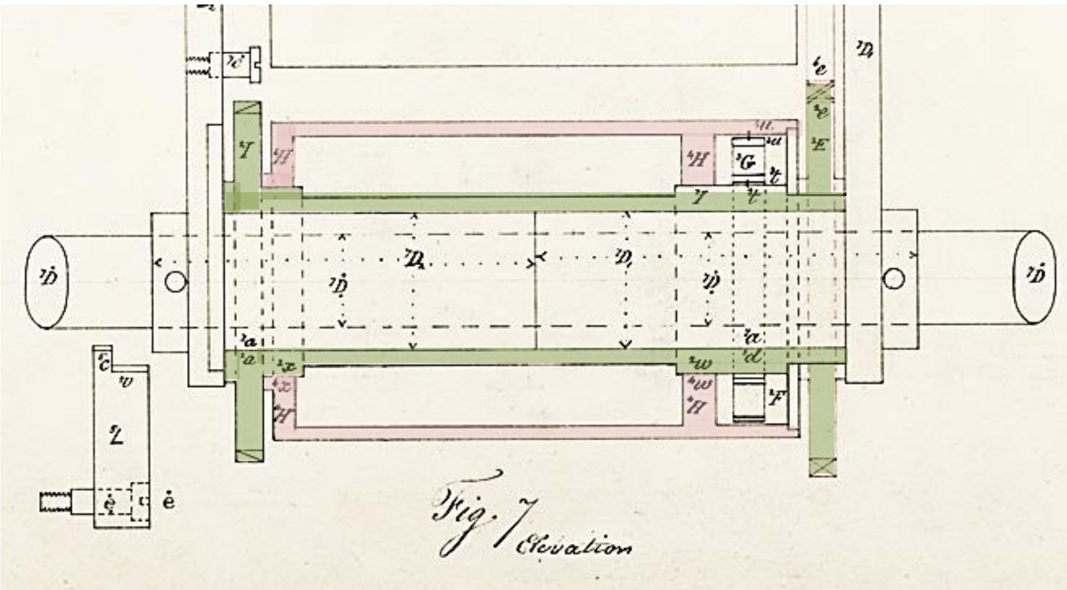


Fig. 5.13: Paper take-up drum assembly (A/165) (colour added).

Each of the two swinging arms, $^7\boldsymbol{D}_1$, $^7\boldsymbol{D}_2$, (A/165, *Fig. 7*) have long bosses pinned at each end (outside the swinging arms) to the shaft $^7\boldsymbol{D}$, the central shaft of the take-up drum assembly. The two bosses extend inside the arms, sleeving the shaft, and meet in the middle. A second assembly, which runs the full width between the two swinging arms, is sleeved over the boss and is free to rotate on the shaft formed by the boss. This second assembly consists of a sleeve, a ratchet wheel ($^2\boldsymbol{I}$, A/165 *Fig. 6*, *Fig. 7*), and gearwheel $^2\boldsymbol{E}$ (*Fig. 7* right). The gear wheel ($^2\boldsymbol{E}$) meshes with a similar gear wheel ($^6\mathscr{E}$) fixed to the feed drum ($^6\mathscr{F}$). The third part of the nested assembly is the barrel of the take-up drum ($^4\boldsymbol{H}$). This thin-walled outer drum rotates on the sleeve which carries the second assembly. Trapped in a recessed annulus between the take-up spool and the sleeve is a clock-spring ($^3\boldsymbol{G}$) (*Fig. 7* right) pinned, at one end, to the inside of the take-up drum, and to the sleeve of the second assembly at the other. The wound clock-spring biases the take-up spool to rotate anticlockwise (A/172, A/165 *Fig. 6*) to maintain positive tension on the paper. A feed pawl ($^5\boldsymbol{L}$) pivots on the question-mark bracket $^3\mathbf{K}$

(A/172) and is biased by a leaf spring ($^{5}\boldsymbol{E}$) to engage with the teeth of the ratchet wheel. The boss on the side with the ratchet wheel (A/165, *Fig. 7* left) has a cam fixed to it (A/165, *Fig. 6*) the outline of which is shown in two positions, dotted and solid.

The final piece of the assembly is a backstop pawl ($^{3}\boldsymbol{I}$, A/165 *Fig. 6*) (also A/172) which engages with the ratchet teeth under pressure from a second leaf spring (A/172, $^{3}\boldsymbol{H}$). The backstop pawl holds the ratchet and gear assembly in fixed relation to the swinging arms during the upstroke. It ensures that the assembly moves as a unit during the printing stroke with no relative motion between the drums.

The solid cam-outline shows the position of the cam with the assembly raised (A/165). During the upstroke the cam disengages the feed pawl ($^{5}\boldsymbol{L}$) by lifting it clear the teeth of the ratchet wheel (solid view in A/165, *Fig. 6).* Towards the end of the return stroke the cam releases the feed pawl to engage with the ratchet wheel (dotted view A/165, *Fig. 6,* and solid in A/172). This halts and holds the ratchet wheel in the position at engagement and for the rest of the stroke the ratchet wheel and gear wheel ($^{2}\boldsymbol{E}$), attached to the ratchet shaft, remain stationary. In returning to the rest position the remaining travel of the swinging arms moves the feed-drum so that its gear wheel ($^{6}\mathscr{E}$) is driven against the stationary gear ($^{2}\boldsymbol{E}$) so as to drive the feed drum clockwise (as in A/172) i.e. the feed drum and stationary gear act as planet and sun gears,. The feed pawl, which is fixed to the stationary framework, and therefore immobile, acts as the leverage point. The backstop ratchet is overridden in this last motion. 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 drum driven by the clock-spring, so tension is maintained.

The action of the pawls, ratchets and gears in arcing up to take an impression and returning to its rest position automatically advances the paper roll by one line-height interval. Tension is maintained by the sprung take-up drum. The outcome is to present, at the end of each print cycle, a fresh area of paper on the roller bar ready for the next impression.

The incremental length of paper wound on 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 during each printing cycle, and therefore the line height, will vary slightly. The direction of the variation will be to reduce line height as the calculation run progresses.

Modifications and Additions

Several details are omitted in the drawings for the paper system. Specifically in A/165 no method is shown of fixing the cam to the boss on the main shaft $^{7}\boldsymbol{D}$, or of fixing the ratchet wheel ($^{2}\boldsymbol{I}$) or gear ($^{2}\boldsymbol{E}$) to the sleeve, and no provision is made for loading the paper stock onto the feed drum. The feed drum is not detachable (A/172, A/173) so loading, using prepared stock on hollow formers, was clearly not intended. The paper needs to be wound onto the feed drum from an external magazine but no means of turning the feed drum is shown. The feed drum is wound anticlockwise to load it. Since the feed drum is geared to the ratchet assembly of the take-up drum, attempting to turn the feed drum anticlockwise drives the mechanism against the feed pawl which blocks motion in the required direction. To freely rotate the drum the feed pawl and back-stop pawl need to be released and there is no provision made for this. Finally, there is no detail given for a method of securing the ends of the paper roll to either the feed drum or to the take-up drum.

Mechanisms for releasing the backstop pawl, loading paper stock from an external magazine, and securing the ends of the paper role, were added to the original design. The general assembly of the revised arrangement is shown in 337 K 24.

Modifications to the Take-up Drum

Several modifications were made to the take-up drum to make provision for omitted features, ease of manufacture, assembly and maintenance. None of the modifications or additions has any implications for the basic design, which is sound.

Securing the paper-roll ends

No provision is made in the original drawings for securing the end of the paper roll to the feed drum or the take-up drum. The preferred method of securing the paper was by a cross-clamp acting as a tenon into an open slot running across the drum (Fig. 5.14). The paper is trapped under the clamp which is fixed at each end by two screws let into cast lugs in the drum. The curvature of the outer surface of the clamp follows the drum 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 drum for balance (for detail see 337 K 481A).

The cross-clamp arrangement for securing the paper to the take-up drum is duplicated on the feed drum 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.

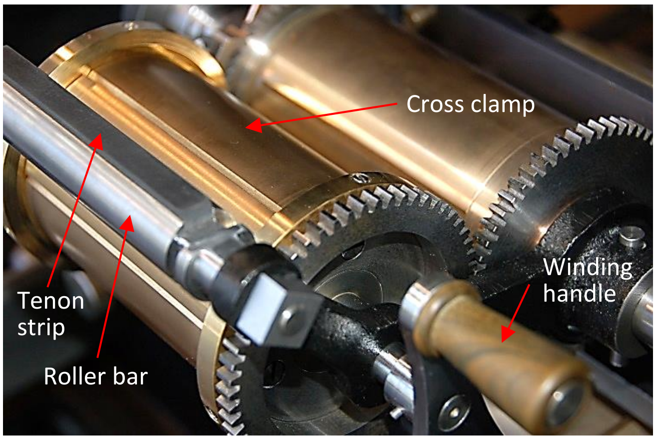


Fig. 5.14: Roller bar and cross-clamp to secure paper ends.

Loading Paper

The drawings make no provision for loading the feed drum. The drawing for the feed drum (A/165, *Fig. 7*) has an inscription “printing paper coiled on this barrel at the commencement” but there is no indication as to how this is to be done. The feed drum has fixed gearing to the ratchet wheel which is itself constrained by two sprung pawls. Loading the feed drum requires counter-clockwise rotation – motion blocked by the feed pawl (A/172, $^{5}\boldsymbol{L}$) which is engaged when the assembly is in the rest position. Turning the feed drum is also resisted by the action of the backstop pawl ($^{3}\boldsymbol{I}$, A/165 *Fig. 6*) (also A/172).

Facilities for paper loading were introduced at the design stage: provision is made for releasing the two pawls, a detachable magazine (Fig. 5.15) for the reserve paper stock from which paper is loaded, and a winding handle is fitted to coil the paper on the feed drum. The general arrangement is shown in 337 K 24 with details of the magazine in 337 K 58.

The detachable magazine consists of two side cheeks separated by three spacers, a loose top roller and two wooden cradles which support the paper roll. 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.

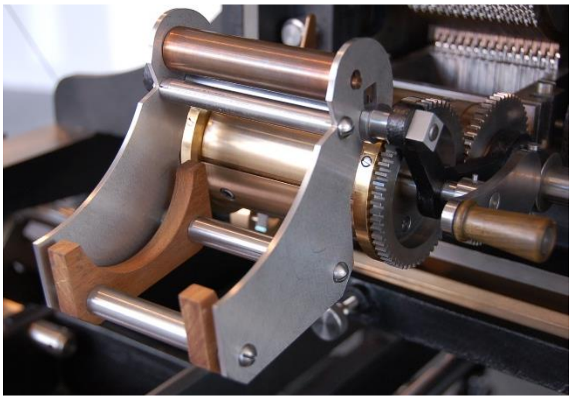


Fig. 5.15: Detachable paper-loading magazine.

The feed drum is wound anticlockwise to load from the magazine. Since the take-up spool is geared to the feed drum, both the backstop pawl and the feed pawl need to be disengaged to allow free rotation of the drums. 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. It may be desirable to rotate the drums at points in the cycle other than when the feed pawl is automatically released by the cam, during paper loading for example, or to clear jams. So a pawl release lever was added (337 K 24, detail K 591, Fig. 5.16). If it is more convenient to halt the cycle with the paper drums in their rest position where the feed pawl is still engaged, the feed pawl can be disengaged and held in its released position by hand while paper is wound from the magazine onto the feed drum (this is the technique described in the **User Manual (2013), 5 Operating the Output Apparatus**, p. 45).

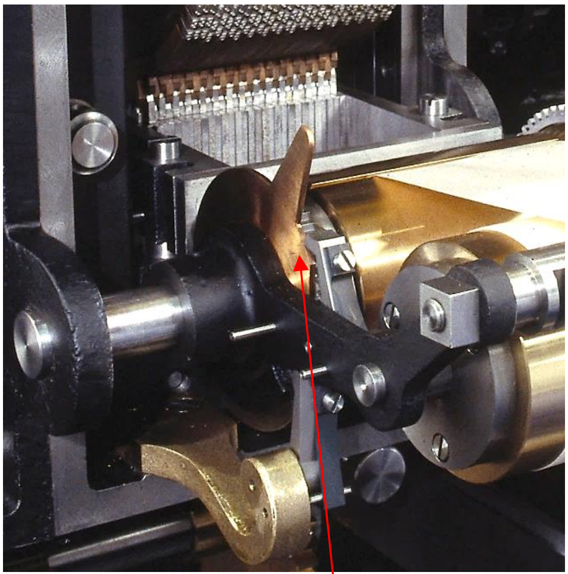


Fig. 5.16: Pawl release lever.

The release lever for the backstop pawl (detail K591) is loosely mounted on the ratchet boss with clearance between the swinging arm and the ratchet wheel. There was space enough to accommodate the lever without modification to the layout. The ramp on the lever (337 K 24) lifts the pawl out of engagement when the lever is operated. The lever is held by a pin in the swinging arm which enters a travelling slot in the lever (K 591). During loading the direction of rotation (clockwise) biases the lever to hold the pawl in the disengaged position.

Paper is wound onto the feed roller from the magazine by turning the handle pinned to the feed drum shaft. The handle is an addition to the original design (Figs. 5.14, 5.15).

The detailed procedure for securing the paper stock and replenishing the paper supply is described and illustrated in the separate **User Manual (2013)**, **5. Operating the Output Apparatus,** pp. 42-46.

Clock-spring chamber

Accommodating the cross-clamp for securing the paper roll end required minor changes in some axial and radial dimensions of the assembly as well as modification to the assembly of the clock-spring chamber. For convenience of machining the mortise for the cross-clamp requires runout off the end of the drum. With the clock-spring chamber integral with the drum, the cross-clamp would form a segment of the chamber outer wall. This would expose the spring when the clamp is removed, for paper stock replenishment, for example, 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 in A/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 drum 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 (337 K 24, 337 K 59).

The take-up drum and the clock-spring chamber are assembled as a sandwich between the two swinging arms which act as side pieces. The gear ($^{2}\boldsymbol{E}$) forms the outer radial wall of the spring chamber instead of the end plate (A/165) which was dispensed with. The clock-spring chamber is driven by a pin let into the drum and protruding into the spring chamber wall. The larger diameter of the pin is secured in the body of the barrel 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", thirty printed digits and 1/4" margin each side, the width of paper roll is 4¼". To allow clamping across the full 4¼" 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 A/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.

Ratchet wheel

The assembly of the ratchet sleeve is not clear from A/165. The ratchet wheel ($^2\boldsymbol{I}$) is shown as one with the sleeve (A/165, *Fig. 7*, left, Fig. 5.13). The ratchet teeth need to be hardened. For ease and economy of manufacture the ratchet wheel is made as a separate item from high-grade hardenable steel and secured to the standard-steel barrel with three screws. 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. The gear ($^2\boldsymbol{E}$, A/165, *Fig. 7*, right) is shown as separate but no means of fixing to the sleeve is shown. $^2\boldsymbol{E}$ is keyed to the ratchet sleeve.

Nested bearings

To allow room within the original outer diameter of the drum for the clamp mortise and for the fixing lugs that secure the paper clamp, the arrangement of nested bearings was modified. The original design shows the outer drum ($^4\boldsymbol{H}$) bearing on the ratchet sleeve ($^2\boldsymbol{I}$), and the full length of the ratchet sleeve is shown bearing on the swinging arm sleeves which are pinned to the shaft ($^7\boldsymbol{D}$) and meet in the middle (A/165, *Fig. 7*, Fig. 5.13). The altered arrangement is shown in 337 K 24. Since there is no real 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. The ratchet sleeve is supported by bearing surfaces at each end only. At the ratchet end the boss is extended through the swinging arm to provide one bearing surface; at the gear end the boss does not extend into the barrel chamber. Instead the second bearing is provided by a phosphor bronze bush (to avoid steel-to-steel bearing surface) in a press fit. Eliminating the two full half-sleeves allows the diameter of the bearing boss at the ratchet end to be reduced from 1¾" to 1⅜". 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 drum assembly.

Capacity of print-roll stock - Explanation

The unloaded diameter of the feed and take-up drums is shown as 3" (A/165, *Fig. 7*). The two views (A/165 and A/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 A/173 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 each time then the effects of the gradual change in the effective diameters, as a result of the reserve stock transferring from the feed roll to the take-up drum, can be ignored.

The housing for the clock-spring (A/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 drum can hold. A further concern is that the tension of the clock-spring may override the backstop pawl and cause unwanted feeds or, in the worst case, runaway. The feed pawl is lifted clear of the ratchet teeth during the upstroke and for most of the return stroke so provides no resistance to runaway or unwanted feeds for most of the print stroke, and the backstop pawl (^{3}***I***, A/165 *Fig. 6*) offers the only resistance to unwanted paper feed during this part of the cycle.

The paper management system revealed 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 drum. This gap is about 0.58" (A/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 be reduced by a factor of four with the added advantage of making loading the paper stock (rolled on manually) more manageable. A radial thickness of 1/8" of paper provides an unbroken run of about twenty-four feet which is a practical length and will serve for an indicative design calculation.

A twenty-four-foot run of paper corresponds to a nominal thirty-one 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 maintain tension. However, the operation of the mechanism shows that the total number of turns of the take-up drum 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 towards the end of the return stroke the feed drum rotates clockwise against the stationary take-up gear to issue a short length of paper (Fig. 5.3). The downward arc of the printing roller during the return stroke wraps this short length of paper around the take-up drum. The essential action transferring paper from the feed drum to the take-up drum is this wrapping motion that takes place towards the end of the return stroke and the take-up drum does not need to turn anything like the number of turns of paper on the feed drum to completely transfer the paper stock from one drum to the other.

At the start of the run, the diameter of the feed roll will be greater than that of the take-up drum. When the distribution of the paper stock is equal, the diameters will be equal; 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 for each result will be slightly longer at the start of the run than that wrapped around the take-up drum – this because the diameter of the paper stock on the feed drum is larger than that of the take-up drum. 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 shorter 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 π x (difference in diameters) = $\pi \div 4 = 0.785$" 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 fifteen turns. The worst-case length discrepancy over the whole 24-foot roll will be 15 x 0.785" = 12" (say). The number of compensatory turns by the take-up spool in either direction will therefore be 12" ÷ (π x 3") = 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 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 a single paper feed with a fresh roll is $\pi D/54 = 0.189$"

where fifty-four 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 twenty-four-foot roll) due to the shrinking diameter of the feed drum as the paper stock depletes is ($2\pi \div 54$) x (thickness of paper stock) = 0.0145". Given that the ratchet advances one tooth per cycle this corresponds to about 8% of the nominal feed for a single line i.e. the difference between line-heights at the start of the print roll and at the end as a result of diameter changes as the paper stock transfers to the take-up drum is 8% in the worst case.

5.4 Locks and Modifications to Printing Racks Lock

The locking technique most widely used in the Engine is the insertion of wedge-shapes to lock, make fine adjustments to centralise a rack or gear wheel, or to detect positionally indeterminate motion (see **3.1 Locks and Security Devices**, p. 29). What was not clear until last was whether wedge locks are also a refined part of the drive mechanism and intended to operate against load i.e. whether their function was confined to ensuring the integrity of digital operation, as in the calculating section, or whether, in the output apparatus, the locks are intended to align the printing and punch wheels to analog standards of exactness sufficient for a typographically acceptable result comparable to hand-typesetting.

Early experiments with the output apparatus showed imperfect alignment of the inked results on the print roll and it was clear that the forces exerted by the locks were not sufficient to drive the stereotyping punch wheels or the printing wheels into alignment, whether or not this was their intended function, and this had been masked by the enduring stiffness in the drive train. The locks were sufficient to hold rack and wheel positions to prevent derangement but not sufficient to effect final registration.

The solution adopted was to modify the drive to one of the locks so that it operates with controlled force of insertion sufficient to align the printing/stereotyping mechanism.

Locks are provided at five separate points in the output apparatus. These operate on:

1. Compound racks meshing with the sectors ($^{5}\mathscr{L}$, A/174, A/176)
2. Printing-wheel racks ($^{7}\boldsymbol{Z}$, A/172)
3. Printing-wheels ($_{n}{}^{9}\mathscr{R}$ A/172, A/173)
4. Stereotyping punch wheels (large) ($^{1}\boldsymbol{I}_{3}$, A/172)

5. Stereotyping punch wheels (***l***$_3$, A/172)

In all five instances locking is by means of a single common wedge-shaped locking bar engaging with an aligned set of V-shaped notches.

At least one lock is needed to force the final alignment and whichever of them is chosen requires modification to apply greater insertion force. The vertical lock on the compound racks at the sector-wheel end is too early in the train: spindle whip, wind-up, and play in the spindle and rack teeth would tend to degrade the settings.

The arms of the printing wheels lock and printing racks lock are relatively slim and were clearly not intended as a drive to operate under significant load in the form as drawn. There is also no obvious way to strengthen the printing-wheels lock. Since the inked copy is anyway primarily for checking, exact alignment on the print-roll is less critical than for the stereotyping punch wheels. However, the stereotyping punch wheels are driven from the printing wheels via horizontal racks ($_{n}{}^{6}\boldsymbol{R}$, A/172) and misalignments in the printing wheels will be transferred. Misalignments persisted after the frictional drag of the printing racks was remedied. If the locks were intended to perform alignment of the print heads to analog standards of exactness then the forces with which they were driven home were inadequate. Experimental stereotyping in Plaster of Paris confirmed this.

Either the design did not provide for sufficiently strong forces on the locks, or the design relied inappropriately on the precision of the drive train to align the heads. In either event the problem of alignment needed solution.

The ideal lock to strengthen would be the crab-claw locks operating on the stereotyping punch wheels as it is the quality of the stereotypes that is the ultimate purpose of the whole apparatus. A crab-claw lock is shown in Fig. 5.17 (A/172 bottom left) for the small punch wheels (left) (the one for the larger punch wheels (right) is not drawn). The claws close simultaneously but only one or the other of the bars engage with the V-notches between the type heads depending on the position of the type wheel. In the position shown in A/172 (Fig. 5.17) the right-hand claw will be active in engaging with the V-notches between the type punches the next time the punch wheels lower. The left-hand locking bar does not engage with the sector gear teeth.

The lock is operated by raising the twin-tyned locking slider ($^{2}\mathscr{Y}$). The angled shoulders of the slider drive the two upper curved followers outwards to close bars (***l***$_3$, ^{1}***l***$_3$) in a rocking action. Lowering the slider operates the lower two curved followers (***l***$^2{}_1$, ^{1}***l***$^2{}_1$) to release the lock. The locking slider rides on a single dovetail slide ($^2\mathscr{Y}$, A/172, A/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 $^2\boldsymbol{A}_{11}$, $^2\boldsymbol{A}_{12}$ (A/173) and contact followers $^3\boldsymbol{W}_1$ and $^3\boldsymbol{W}_2$ (A/172, A/173).

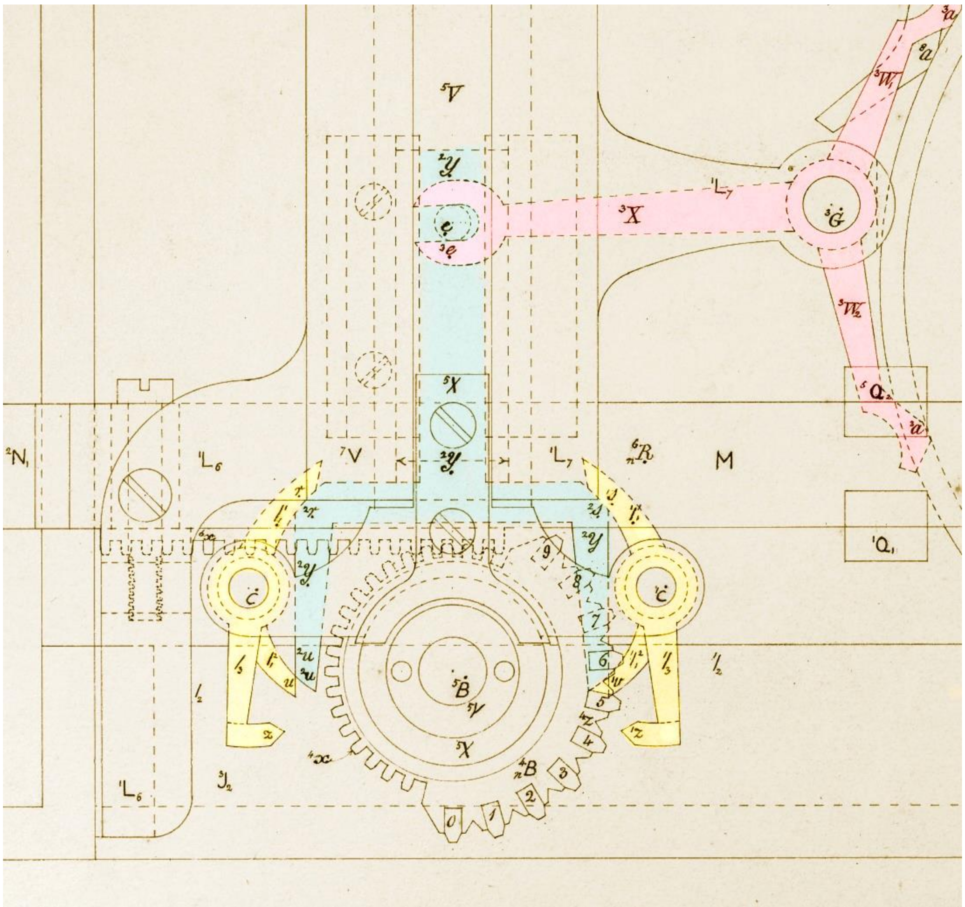


Fig. 5.17: Crab-claw lock for stereotyping punch wheels (A/172, colour added).

The claw-lock is not redrawn for the large stereotyping punch wheels (A/172 right) though it is clear from the provision of the dovetail slide, cams and followers that the method of locking the large stereotyping wheels is a repeat of their smaller counterparts. The cams in this case are $^2\boldsymbol{A}_{13}$ and $^2\boldsymbol{A}_{14}$ and the followers, $^1\boldsymbol{W}_1$, $^1\boldsymbol{W}_2$ (A/173).

However, there is no obvious way of increasing the operating force of the crab-claw locks to force better alignment of the punch wheels (see **4. Output Apparatus**, **Implications of Rack Offset**, p. 61 for related discussion).

By elimination, the lock selected for strengthening was the printing-racks lock ($^5\mathscr{T}_2$ **,** A/172 centre). There is space in the original design for an additional mechanism without modifying the layout of the shafts and axes, and a way was found of deriving the driving forces needed from existing mechanism.

Modified Printing Rack Lock

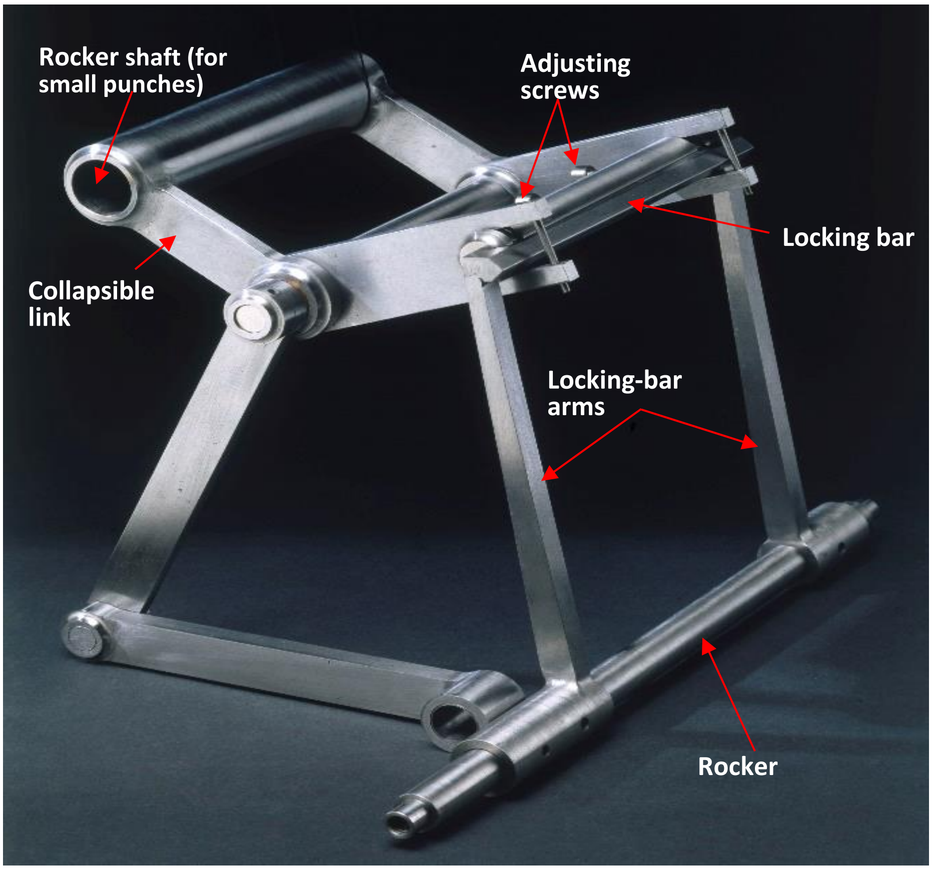


Fig. 5.18: Modified printing racks lock drive.

In the original arrangement a T-bar cam ($^{2}\boldsymbol{A}_{6}$) is shown at 6-o'clock in A/175 left. The follower arm and locking bar ($^{5}\mathscr{T}_{2}$) are shown in elevation in A/172, A/175, and in plan on A/173 (the locking bar is obscured in the plan view and is shown in dotted outline.) The T-bar acts on the contact follower ($^{5}\mathscr{S}$) to drive the lock into engagement. The lock disengages by dropping out after the T-bar cam has passed, or being kicked out when the rack next moves.

In the modified scheme (Fig. 5.18) the wedge-shaped locking bar and side-arms ($^{5}\mathscr{T}_{2}$) are retained and a collapsible-link drive was added to provide positive controlled force to drive the locking bar home. The additional (sturdier) mechanism appears on the left of the vertical centre-line in Fig. 5.18. The wedged locking bar, locking-bar arms, and shaft are retained as originally drawn and the locus of the locking bar is still determined by the length of the original arms ($^{5}\mathscr{T}_2$, $^{5}\mathscr{T}_1$ A/172, A/173). The new mechanism is pivoted on two existing fixed centres: the rocker shaft ($^{5}\mathscr{A}$) of the locking-bar arms (Fig. 5.18 bottom right, A/172 bottom right, A/173), and the rocker shaft ($^{8}\boldsymbol{I}$) for the small stereotyping punch wheels (Fig. 5.18 top left, A/172 near top left, A/173). Both fixed centres are part of the original design. The two additional pivots for the collapsible links have moving centres.

In the original design the cam follower ($^{5}\mathscr{S}$) drives the locking-bar arms via shaft $^{5}\mathscr{A}$ to which the arms are fixed (A/172). In the modified scheme the locking-bar arms are free to rotate on the shaft and the locking-bar is instead driven by the collapsible link mechanism. The lower horizontal drive links (Fig. 5.18) are fixed to the shaft (the second lower horizontal drive link is obscured in shadow in Fig. 5.18). The lock engages when the rocker shaft ($^{5}\mathscr{A}$) turns clockwise driven by cam follower $^{5}\mathscr{S}$. The lowermost link of the collapsible link assembly (horizontal in Fig. 5.18), which is fixed to the shaft, is driven clockwise and this drives the connecting link upwards, driving the collapsed link into its straightened position. The mechanical advantage is substantial and the locking bar is firmly driven home against frictional resistance in the racks and rack drive train. The lock engagement is sustained by the pressure of the T-bar cam (A/175) acting on the follower $^{5}\mathscr{S}$. Pressure is released when the T-bar cam has passed and the lock disengages by dropping out. The links collapse and the mechanism reverts to the fallback state. The cams and timing remain unaltered.

The locking bar is sheathed in a loose sleeve slotted into the drive links (Fig. 5.18). Adjusting screws were fitted in the back of the sleeve that holds the locking bar. These allow micro-adjustment of the insertion depth into the V-notches of the racks to ensure that the apex of the wedge is pressed fully home when the lock is active, and also to align the locking bar for even pressure in the V-grooves along the run of printing racks. Additional field modifications were made to the mechanism in Fig. 5.18. Increasing the

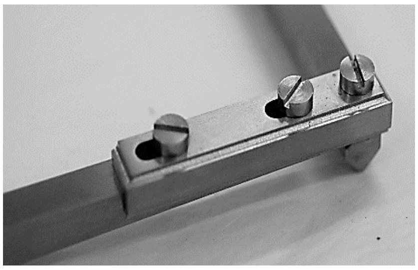


Fig. 5.19: Splint (detail).

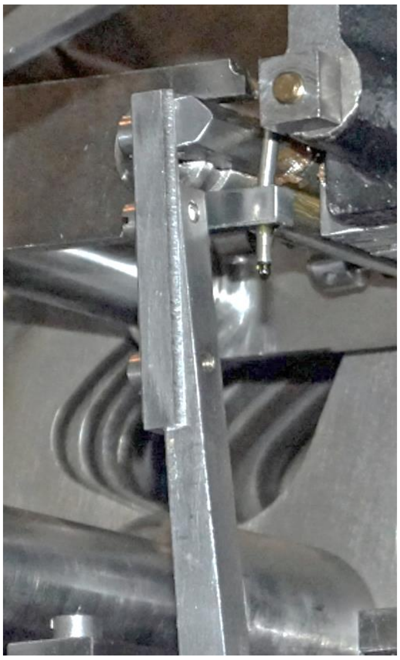


Fig. 5.20: Splinted locking bar arms.

insertion depth using the adjusting screws in the sleeve drove the locking bar past the limits of the retaining pillars and deformed them. The top fixing points of the retaining pillars were removed to allow the pillars to spring forwards and the locking bar is sandwiched between the sprung pillars and the adjusting screws. A final modification was to splint the locking bar arms (Figs. 5.19, 5.20) to allow adjustment of the effective extension of the arms again for accurate alignment. Adjustment is made using screws and slotted holes in the splint which wraps around the back of the arms (Fig. 5.19).

5.5 Cams, Drive and Control

The operation of the inking, printing and stereotyping mechanisms is co-ordinated and controlled by a set of fourteen vertical cams located inside the printing apparatus frame (A/173). The cam cluster consists of fourteen cams of which ten are conjugate pairs i.e. two cams that are geometric inversions of each other – where there is a rise on one, there is a corresponding fall on the other so that the cam followers provide positive drive in both directions.

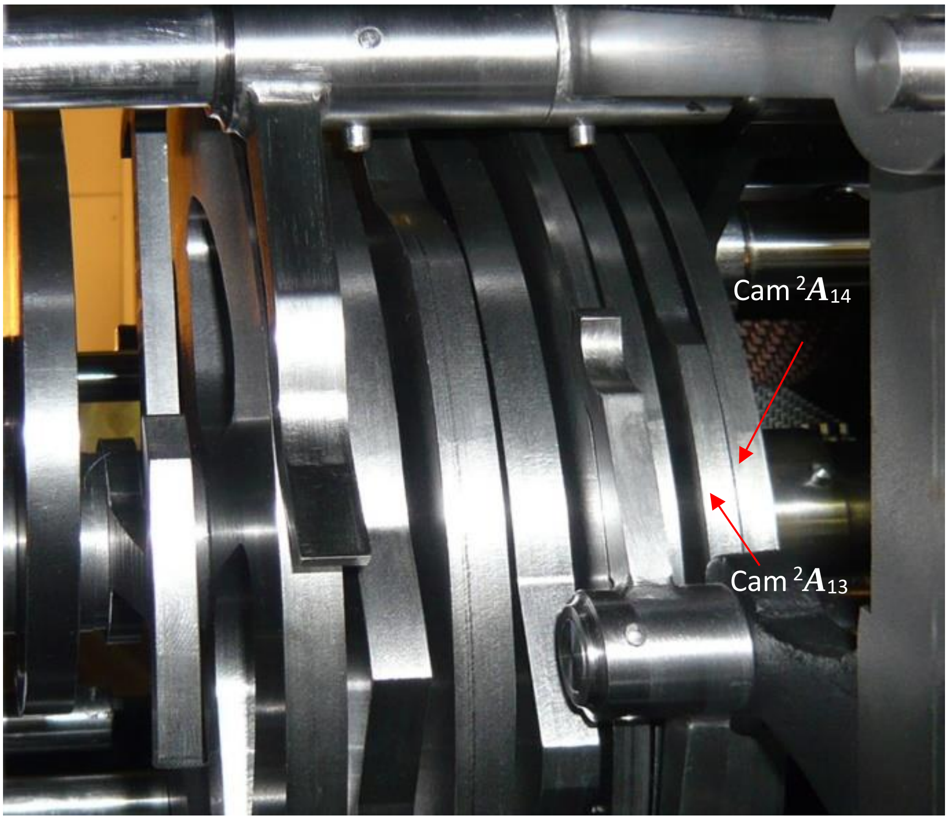


Fig. 5.21: Output apparatus cams (part).

The set of cams is shown in plan in A/173; partial elevations and cam profiles are given in A/172 and A/175. The cam shaft (${}^{2}\boldsymbol{D}$ A/172, A/173) is driven by gear wheel, ${}^{2}\boldsymbol{A}_{1}$ (A/173) from the main drive shaft, ${}^{4}\mathscr{C}$ (A/163) via an idler (${}^{6}\boldsymbol{D}$, A/163). The fourteen cams are annotated ${}^{2}\boldsymbol{A}_{2}$ through ${}^{2}\boldsymbol{A}_{16}$ (A/173) (${}^{2}\boldsymbol{A}_{15}$ is not part of the cam sequence).

The cams and the mechanisms they control are as follows:

Cam (A/173)	**Output Apparatus Cams Mechanism Controlled**
${}^{2}\boldsymbol{A}_{2}$ ${}^{2}\boldsymbol{A}_{3}$	Compound-racks lock (A/172, A/174, A/175)
${}^{2}\boldsymbol{A}_{4}$	Printing stroke drive (A/172, A/175)
${}^{2}\boldsymbol{A}_{5}$	Printing-wheels lock (A/175)
${}^{2}\boldsymbol{A}_{6}$	Vertical (Printing)-racks lock (A/175)
${}^{2}\boldsymbol{A}_{7}$ ${}^{2}\boldsymbol{A}_{8}$	Stereotyping punch wheels (small) (A/172, A/173)
${}^{2}\boldsymbol{A}_{9}$ ${}^{2}\boldsymbol{A}_{10}$	Stereotyping punch wheels (large) (A/172, A/173)
${}^{2}\boldsymbol{A}_{11}$, ${}^{2}\boldsymbol{A}_{12}$	Stereotyping punch wheels lock (small) (A/172, A/173)
${}^{2}\boldsymbol{A}_{13}$ ${}^{2}\boldsymbol{A}_{14}$	Stereotyping punch wheels lock (large) (A/172, A/173)
${}^{2}\boldsymbol{A}_{16}$	Inking roller (A/172, A/173 bottom)

Compound-racks Lock

The operation of the compound-racks lock is described above (see **Locking the Compound Racks**, p.57).

Printing Stroke Drive

The linkage driving the printing assembly is shown in elevation in A/172. The single heart-shaped cam, ${}^{2}\boldsymbol{A}_{4}$, shown in elevation A/175, and in plan as part of the cam stack towards the top of A/173 (but omitted in A/172). The roller follower ($\boldsymbol{H}^{1}$ A/172) pivots on rocker shaft ${}^{4}\boldsymbol{B}$ and drives fixed-angle link $\boldsymbol{H}^{2}$. 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 counterweight, ${}^{7}\boldsymbol{C}^{2}$ (bottom right A/172). The counterweight initiates the return stroke which is controlled by the pressure of the follower on the retreating profile of the cam.

Printing-wheels Lock

The printing-wheel lock cam ($^2\boldsymbol{A}_5$) is the single T-bar cam shown at about 3-o'clock on A/175 right with the follower arm and wedge-shaped locking bar top right of the same view. The follower ($^8\boldsymbol{P}$) and locking bar ($^8\boldsymbol{Q}_2$) are duplicated in A/172 but without the T-bar cam.

The T-bar strikes the follower arm and drives the live rocker shaft $^8\boldsymbol{L}$ (A/172, A/173) to which it is pinned. The locking bar assembly, $^8\boldsymbol{Q}_2$, also keyed to the rocker shaft, is driven so as to drive the locking bar into the run of notches to immobilise the print wheels. The drive is unidirectional i.e. the locking action stops once the T-bar has passed. No provision is made for the locking bar to clear the V-notches after the locking window is over. The locking bar rests under its own weight on the printing wheels when not actively locking and is bounced about by the V-notches as the printing wheels rotate.

Printing-racks Lock

The locking bars ($^5\mathcal{T}_{\mathbf{2}}$, $^5\mathcal{T}_{\mathbf{1}}$ A/172, A/173) for the vertical printing racks ($^7\boldsymbol{Z}$) and for print wheels ($_n{}^9\mathcal{R}$) are operated by pivoted levers directly driven by cam followers.

The drive, operation and modification of the printing racks lock is described in **5.4 Locks and Modifications to Printing Racks Lock**, p. 91.

Stereotyping Punch Wheels

The small stereotyping punch wheels ($_n{}^4\boldsymbol{B}$, A/172) are raised and lowered by conjugate cam pair $^2\boldsymbol{A}_7$, $^2\boldsymbol{A}_8$ (A/173) via cam followers $^8\boldsymbol{D}_1$, $^8\boldsymbol{D}_2$ shown in elevation in A/172. Bosses for the two forked end levers ($^8\boldsymbol{D}_3$) are pinned to the live rocker shaft ($^8\boldsymbol{I}$). The forked-end lever raises and lower the stereotyping wheel assembly which runs in fixed dovetail slides made up of moving slider $^5\boldsymbol{V}$ set in framing slides $^1\mathbf{L}_7$ and $^7\mathbf{V}$**.**

The arrangement is duplicated for raising and lowering the large stereotyping punch wheels. The conjugate cam pair in this case is $^2\boldsymbol{A}_9$ and $^2\boldsymbol{A}_{10}$. Cam profiles are shown in A/175 (left) and cam followers are shown in A/175 and A/172. The forked end lever ($^3\boldsymbol{B}_3$) drives the assembly in dovetail guides as for the small stereotyping wheels.

Stereotyping Punch-wheels Lock

The twin-tyned locking slider (A/172, Fig. 5.17) rides on a single dovetail slide (${}^{2}\mathcal{Y}$, A/172, A/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 ${}^{2}\boldsymbol{A}_{11}$, ${}^{2}\boldsymbol{A}_{12}$ (A/173) and contact followers ${}^{3}\boldsymbol{W}_{1}$ and ${}^{3}\boldsymbol{W}_{2}$ (A/172, A/173).

The operation of the claw-lock is described in more detail in **5.4 Locks and Modifications to Printing Racks Lock**, p. 91.

The claw-lock for the small punch wheels is not redrawn for the large punch wheels though it is clear from the provision of the dovetail slider, cams and followers that the method of locking the large punch wheels is a repeat of their smaller counterparts. The cams in this case are ${}^{2}\boldsymbol{A}_{13}$ and ${}^{2}\boldsymbol{A}_{14}$ and the followers ${}^{3}\boldsymbol{B}_{1}$, ${}^{3}\boldsymbol{B}_{2}$.

Inking Roller

The inking roller cam (${}^{2}\boldsymbol{A}_{16}$), follower (${}^{4}\mathscr{U}_{2}$), drive links (${}^{4}\mathscr{U}_{3}$, $\mathscr{D}$) are shown in plan in A/173 bottom centre and in elevation in A/172. The cam arm sweeps against the scythe-shaped contact follower which rocks on the fixed pivot ($\mathcal{K}$) working against the action of the compression spring, ${}^{1}\boldsymbol{F}$ (A/172). A long link ($\mathscr{D}$) operates link ${}^{1}\boldsymbol{K}_{3}$, which pivots on fixed shaft ${}^{1}\boldsymbol{K}$ to drive the two swinging arms, ${}^{1}\boldsymbol{K}_{2}$ and ${}^{1}\boldsymbol{K}_{1}$. The travel of the inking roller towards the print heads is gradual while 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, and it restores the inking roller after the downward sweep to the type heads.

6. Stereotyping

The output apparatus produces printed inked results on a paper roll as described in Chapter 5, as well as automatically stereotyping results i.e. impressing results into suitable material contained in two trays. The stereotypes were intended to be used as moulds for making printing plates for use in conventional printing presses of the time (Figs. 2.5, 2.7, 4.1).

This section describes the design, operation and control of the travelling platform that automatically repositions the two trays to receive results from punch wheels when, set with each new result, they lower to make impressions on soft material in the trays below.

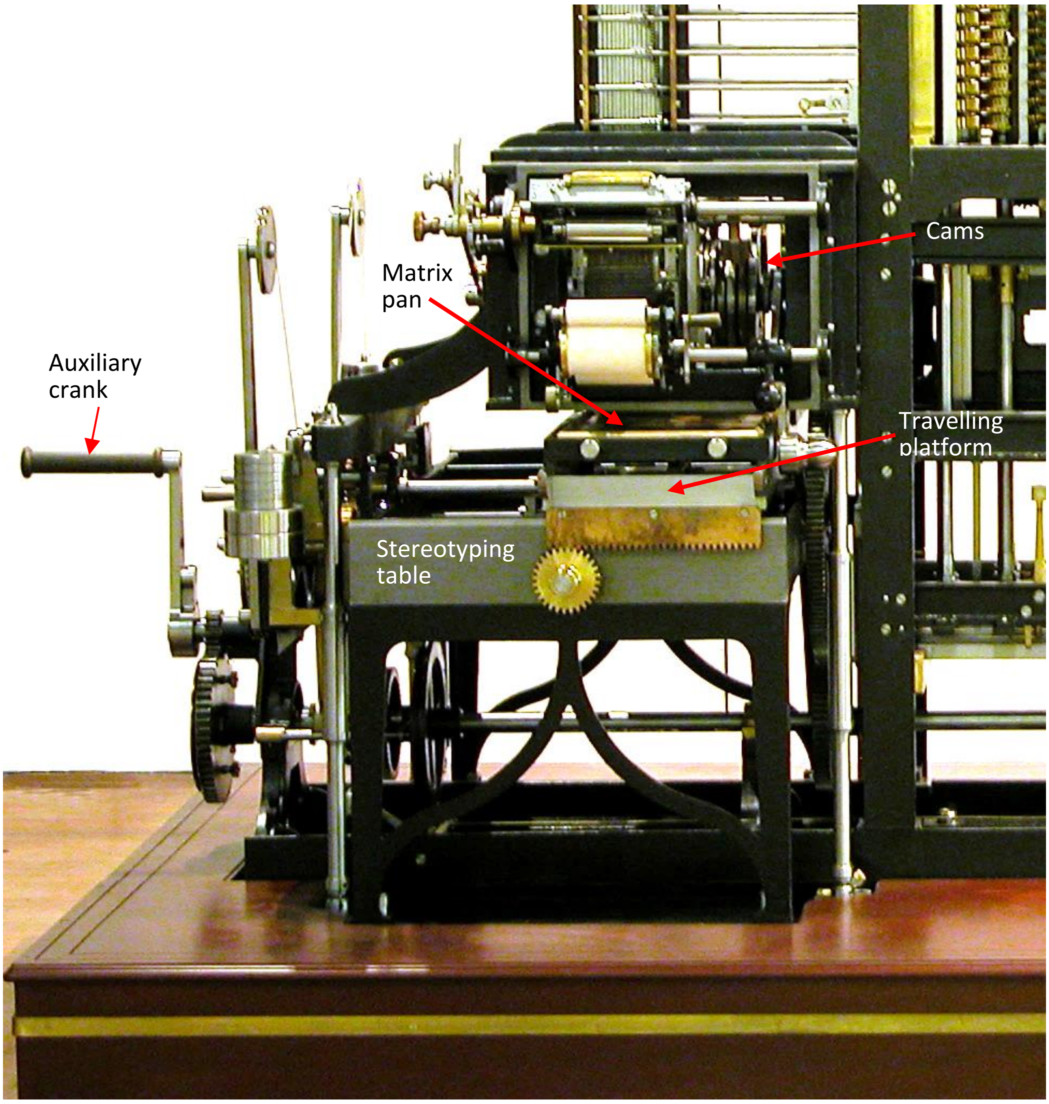


Fig. 6.1: Output Apparatus, Elevation.

The printing and stereotyping apparatus is shown in A/163 left (Figs. 2.2, 2.3, 2.5, 6.1). The printing apparatus is the box-like assembly (Fig. 2.3) bolted to the left-hand vertical frame members roughly half way up the engine and includes a local set of cams that drive and orchestrate the motions required for both printing and stereotyping. The stereotyping table is the apparatus immediately below with the signature wish-bone struts that are part of the cast legs ($^{1}\mathbf{A}$, A/163, Fig. 6.1). The stereotyping apparatus is wider front to back than the calculating section (A/164, Fig. 2.2 Plan) and the cast stand for the stereotyping table rests on the base plinth not on the narrower rails on which the rest of the Engine rests i.e. the cast legs of the stereotyping table fall outside the main base rails (Fig. 2.5).

The transfer of thirty-digit results from the results column in the calculating section, to the print wheels for hardcopy printing, and to the punch wheels for stereotyping, is illustrated in Fig. 4.2 and described in sections **4. Output Apparatus** and **5. Printing**.

The thirty-digit result of each calculating cycle is set up at the same time on both the small and large punches each of which have thirty type wheels. The punch wheels are shown in A/172 and in outline at the top of A/147. The two sets of punch wheels are lowered together to impress the result into the material in the trays below to provide two sizes of impression, large and small, in the same action. There is missing detail in the original design drawings and it is unclear whether the large and small pans were intended to receive impressions at the same time or whether one of the two formats excluded the other (see **Split Carriage**, p. 106).

The horizontal racks transfer results from the print wheels to both the sets of small and large stereotyping punch wheels (Fig. 4.2). The racks (${}_{n}{}^{6}\boldsymbol{R}$) shown in End View A/172 and as curved 'push rods' (${}_{n}{}^{7}\boldsymbol{R}$) in Plan View A/173, narrow from 1/8th-inch pitch for the large punch wheels down to 1/16th- inch pitch for the small wheels (Fig. 6.2). The overall structure of the rack assembly is that of a rectangular frame made up of cast framing members bolted together. The frame is formed at the narrow end (A/173 left) by framing piece $^{2}\mathbf{N}_{1,}$ at the broad end by $^{3}\mathbf{N}_{2}$ (A/173 right) and by long side pieces, $\mathbf{M}_1$ and $\mathbf{M}_2$ (A/173 top and bottom respectively). The racks are boxed in at the full width end by the main side piece framing members ($\mathbf{M}_1$, $\mathbf{M}_2$). At the narrow end the straight sections are sandwiched between two edge blocks. The curved sections of the racks are formed in the shapes shown and retain their curvature when relaxed. They do not rely on lateral contact to retain their form and the curvature is not a sprung form resulting from lengthwise compression. Rather, the shapes are machined forms that are retained with the racks in a dismantled state (337 K 391 A-D).

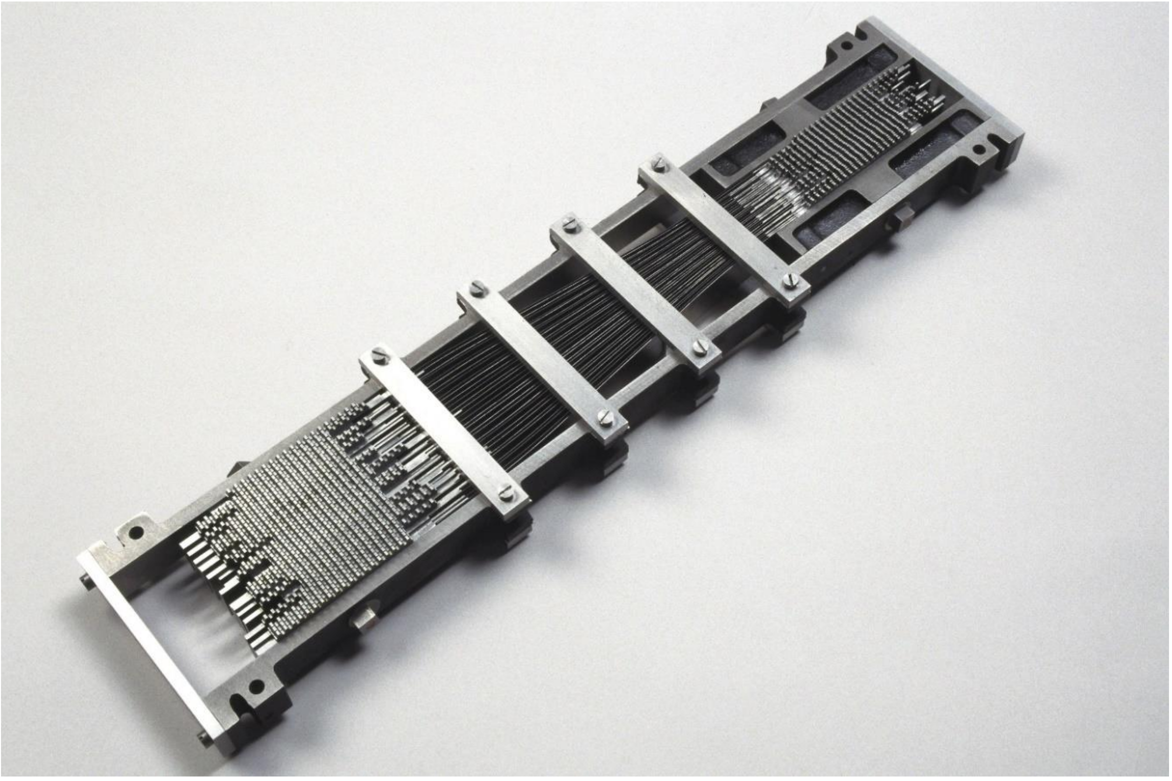


Fig. 6.2: Horizontal racks in mounting frame.

The stereotyping sectors (punch wheels, Fig. 4.3) have vertical and rotational degrees of freedom only and have no freedom to move in the horizontal plane. The position of the result on the page is determined not by lateral motion of the heads but by the position of the tray below the heads. The stereotyping apparatus is in effect a programmable X–Y platform that repositions the trays under the stereotyping heads anew for each calculating cycle to receive each new result.

The format of the tabulated results in the stereotyping trays is programmable. The apparatus provides options for line-to-line results (down the page) with automatic column wrap to the top of the next column, column-to-column results (across the page) with automatic line-wrap at the end of each line, and options for results to appear in single or multiple columns. The number of lines per page, and leaving blank lines between groups of lines for ease of reading, are alterable, as are the page margin-widths and spacing between columns.

The various layout options are selected by choosing pattern wheels preloaded onto the apparatus (Fig. 2.5). The pattern wheels ($^{3}\boldsymbol{N}$, ^{6}N) are shown in A/163 (elevation), A/164 extreme left (plan), and in A/166. One pattern wheel ($^{3}\boldsymbol{N}$), selected from a possible choice of four, determines line-to-line spacing (right-hand set-in Fig. 2.5); a second pattern wheel (^{6}N) (left-hand set in Fig. 2.5), also selected from a possible set of four, controls column-to-column movement. Selection is made by means of a sliding pawl the position of which determines which of the pre-mounted set of pattern wheels is active (see p. 117).

The formatting features are not independently variable. The pitch and arrangement of teeth on a single line-to-line pattern wheel, for example, determines a fixed and unalterable combination of line features: line-height, how many results (if any) are grouped before a blank line is left, the height of the blank line, how many lines per page, top and bottom page margins. The apparatus has capacity for up to four pre-loaded pattern wheels each of which offers a fixed combination of formatting features. Pattern wheels for different formats can be mounted as required by demounting one or more of those already mounted. A similar set of four fixed combinations of column-to-column features are provided as standard: how many columns per page, page margins, and margins between columns. Any of four of the line-to-line pattern wheels can be selected with any of the column-to-column pattern wheels i.e. the line-to-line pre-set combination options are independent of the column-to-column options. (For page formatting and layout options see 337 X 23).

The stereotyping mechanism forms part of the control mechanism of the calculating sequence. When a page is completed in the format pre-set by the pattern wheels, a cylindrical weight is released into a trough (***B*** A/163 bottom left). The trough operates the trip lever connected, via a cable (ℓ) guided by pulleys, to the main-drive clutch above the cam stack at the far right of the engine (A/163). When the end-of-page condition trips the lever the clutch at the main cam stack disengages the drive and the handle runs free. Uncoupling of the drive at end of page halts the engine without any overrun and ensures the integrity of the figure wheel settings and control mechanism when the calculation is resumed so that the run of results is continuous. Automatic halting at the end of a page allows the material in the trays to be renewed. This is done by fixing fresh trays, prepared with suitable material, on the movable carriage.

Main Drawings: A/147, A/163, A/164, A/166, A/172, 337 X 23)

6.1 Travelling Platform

The material receiving the impressions from the stereotyping punch wheels is held in shallow trays called, in the original drawings, 'matrix pans' (^{1}**I**, A/147 top, Fig. 6.1) and separate matrix pans for large and small type are shown. The matrix pans rest in a travelling frame or carriage ^{1}**H** (A/147 top). The soft material, wet Plaster of Paris, for example, needs to set before being removed from the pan or before being used as a mould while still in the pan. The pans therefore need to be removable to allow replacements, pre-loaded with soft material of the correct consistency and depth, to be mounted on the travelling frame so that tabulation can proceed without waiting for the material to set before a tray can be freed up.

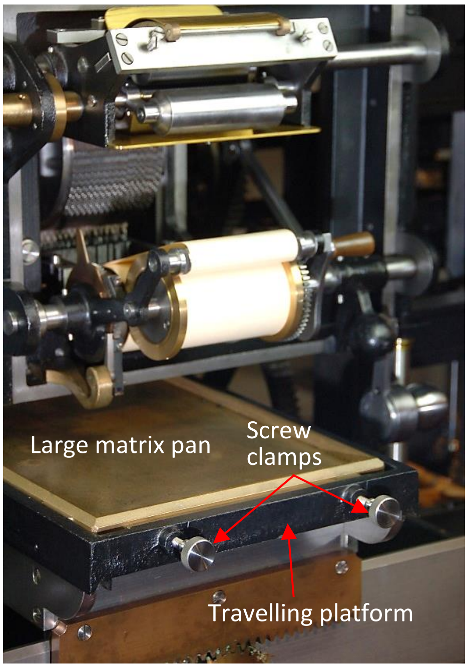


Fig. 6.3: Matrix pan and travelling platform.

No means of fixing the pans to the carriage is shown, nor how their separation on the carriage is retained. The Elevation (A/147 top left) shows a flush fit between the pan and the carriage frame along the left and right edges, and the End View shows the same clearance to the punch wheels for both small and large pans i.e. the two pans are shown to be in the same horizontal plane. The position of the pans on the carriage appears to rely on a tight fit left-to-right in the width of the frame, with the pans wedged in place along their length i.e. wedged front-to-rear as viewed from the front of the Engine. Registration by wedging would be consistent with contemporary printing practice though the drawings are not explicit in this respect.

The carriage bearing the pans runs on two pairs of inverted V-shaped runners. One pair (^{2}**G,** ^{1}**H,** A/147 Elevation) gives the freedom to traverse line-to-line down the page in any given column; the other pair (**F**, ^{2}**G,** A/147 End View) at right angles to the first, gives column-to-column travel across the page on any given line i.e. the orthogonal sets of sliders provide the two degrees of freedom required for independent X–Y motion.

Line-to-line motion of the pans corresponds to right-to-left motion of the carriage in the End View (147 top right) (the punch wheels are shown poised above the top-of-page position for both trays, A/147 End View). 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 (A/147 top left) i.e. the carriage traverses right to left when progressing from the first column to the next column along.

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 A/147.

The Plan View in A/147 shows the two gears ($^5\boldsymbol{S}$, $^5\boldsymbol{P}$ engaging with two racks $^1\mathbf{R}_2$ and $^1\mathbf{R}_1$ respectively which would be fixed to the underside of the carriage (Fig. 6.5). Rotation of the shaft ($^7\boldsymbol{A}$) advances the platform line-to-line. When stereotyping with small type, the carriage is driven by a single pinion $^5\boldsymbol{R}$ 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 A/147 or A/163.

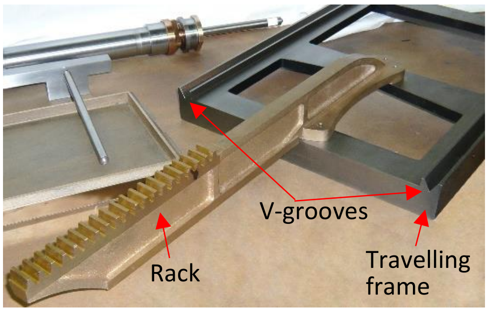


Fig. 6.4: Small pan frame and rack (inverted)

The pinions ($^5\boldsymbol{S}$, $^5\boldsymbol{P}$) for the double rack, and the pinion ($^5\boldsymbol{R}$) for the single rack ($^4\mathbf{R}$) are driven by the same drive shaft, $^7\boldsymbol{A}$. Since the carriage for both matrix pans is shown as a single assembly the drives as shown would be mutually exclusive: if both were engaged there would be a contention between the different pitches of the single pinion drive and the double pinion drive. It would seem then that stereotyping in both small and large type simultaneously was not intended, and only one type size used at a time with the drive for the unused size disengaged. Again, there is no indication of how one of the drives is to be disabled.

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 suggests that having 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, provide the option of up to three-column layouts compared to a maximum of two columns in the larger type.

The pinion, $^5\boldsymbol{R}$, is shown centred between the side frame members though the detail of what is fixed to what is unclear (see below). A travelling keyway is shown along the length of the shaft $^7\boldsymbol{A}$ (A/147 Plan) which allows the whole assembly to traverse during column-to-column motion (top-to-bottom motion in A/147 Plan). The column-to-column traverse is best seen in A/163 left and A/147 Elevation). (The pinion is shown wider in A/163 than in the two views in A/147. This is a drafting inconsistency).

Split Carriage

In the reading above the three views in A/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; right-to-left motion in A/147, End View); the long runners track on transverse runners to enable column-to-column movement (left-to-right facing the engine in A/147, Elevation). The larger line-pitch required for the larger typeface is provided by the pair of racks $^{1}\mathbf{R}_2$ and $^{1}\mathbf{R}_1$ engaging with pinions $^{5}\boldsymbol{S}$, $^{5}\boldsymbol{P}$ 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 $^{5}\boldsymbol{a}$).

A central issue is that the single travelling platform bearing the two matrix pans has two line-to-line drives that have different feed rates and the drives will be in damaging contention if they operate at the same time.

A clue to the original design intention may be to resolve the issue of how the two sets of pinions were to be driven as there may be provision for disabling one of the drives so that only one of the two matrix pans operated at a time. A/147 Plan shows an uninterrupted sleeve $^{5}\boldsymbol{B}$ on the drive shaft $^{7}\boldsymbol{A}$. The sleeve appears to be keyed to the shaft by a sliding key along the full length of the shaft (dotted lines in A/147 Plan) – this to provide line-to-line drive independent of the column-to-column traverse. The End View shows a key (e) cut into the shaft. It is unclear (depending on where the section is considered to be taken) whether the pinions $^{5}\boldsymbol{S}$, $^{5}\boldsymbol{P}$, or the sleeve, are keyed to the shaft. If the pinions are keyed to the shaft then the sleeve can act as a spacer separating the pinions which are sandwiched by collars against the sleeve; 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.

An additional more minor omission is that no means is shown of fixing or securing the single small rack ($^{4}\mathbf{R}$) to the carriage. It is drawn levitating freely in A/147 End View.

From the analysis of the drive trains as drawn there is no evidence of provision for disabling one of the drive mechanisms to avoid contention between the two drives. Access to the underside of the carriage to disengage either of the drives is anyway restricted and it is difficult to see how a convenient means of disengaging the drives by disconnection would be devised. A further point against disconnection is that it is essential that the racks are replaced in exactly the same mesh so as to ensure correct positioning under the stereotyping heads as the pinions are phase-locked to the pattern wheels. If the racks are incorrectly remeshed, the punch wheels risk fouling the walls of the pans and results would anyway be displaced on the page. Refitting the racks exactly is an operational constraint. The flanges on either side of the central pinion ${}^5\boldsymbol{R}$ may be intended to act as guides if the rack is to be uncoupled and refitted – a marginal point.

If the pans are secured in the positions as in A/147 End View then, when stereotyping with the large pan, the small punch wheels will foul the trailing wall of the small pan (at the end-of-page) as the carriage advances down the page. Initially this supported the idea that simultaneous stereotyping using both large and small types was not intended but the conclusion is based on the premise that the two pans are fixed to the carriage.

The Mechanical Notation was used to in an attempt to clarify the carriage drive. The sleeve ${}^5\boldsymbol{B}$ has the same index of identity (=5) as the pinion ${}^5\boldsymbol{R}$ which indicates that they are parts of the same piece and act as one. The Notation shows that the central pinion ${}^5\boldsymbol{R}$, as well as both pinions ${}^5\boldsymbol{S}$, ${}^5\boldsymbol{P}$ are driven by ${}^5\boldsymbol{B}$ which the notation shows as loose on shaft ${}^7\boldsymbol{A}$.

The conclusion is that ${}^5\boldsymbol{B}$ is integral with all three pinions, ${}^5\boldsymbol{R}$, ${}^5\boldsymbol{S}$, and ${}^5\boldsymbol{P}$, and that ${}^5\boldsymbol{B}$ is driven rotationally by shaft ${}^7\boldsymbol{A}$ to which it is keyed, to produce linear motion of the carriage, and that ${}^5\boldsymbol{B}$, ${}^5\boldsymbol{R}$, ${}^5\boldsymbol{S}$, and ${}^5\boldsymbol{P}$ make up a single assembly free to move axially along shaft ${}^7\boldsymbol{A}$ sliding on the key for column-to-column motion.

A resolution to the difficulty (contention of the two drives) is suggested by the Notations which indicate that the small pan might be free to slide within the frame of the moving carriage and that the two drives can therefore operate without conflict at the same time. The small pan is identified in A/147 Plan and End View as ${}^7\mathbf{K}$**.** In the notational convention devised by Babbage underscoring a part-identifying character by a single line denotes 'Linear Motion in Plan'.[1] ${}^7\mathbf{K}$ and the rack that drives it, ${}^4\mathbf{R}$, are underscored three times in A/147 Plan while the large-pan identifier, ${}^1\mathbf{I}$, and drive racks, ${}^1\mathbf{R}_2$ and ${}^1\mathbf{R}_1$, are underscored twice. This would indicate that the small rack has an additional degree of linear freedom in the horizontal plane i.e. that the small pan is driven by the central rack and pinion and moves relative to the large pan which itself is moving. The small pan would thus have three degrees of horizontal freedom, and the large pan, two. An additional supportive indicator is found in A/178/4 (bottom right), the Trains notation for the frames, which shows the small

[1] Bromley, Allan G, ed. *Babbage's Calculating Engines: A Collection of Papers by Henry Prevost Babbage*. Vol. 2. Los Angeles: Tomash, 1982. 'Alphabet of Motion' symbol table, June 1851, pp. 250-1.

pan driven by both the carriage frame as well as the central rack and pinion. (A/178/5 also shows **K** with three underscores in the section labelled 'Advance from Line to Line' for the small pan).

The interpretation from the Notation is suggestive rather than conclusive as there are puzzlements about notational consistency. For example, rack ^{4}**R**, and the other racks, are shown with upright characters which goes against the convention of moving part identifiers being italicised. This could be a generic issue in the Notation as to how a part, moving in relation to a part to which it is connected and is itself moving, is annotated. More significantly the small pan is identified as ^{7}**K** in A/147 End View and Plan i.e. the pan has a different index of identity to that for the rack and this would ordinarily indicate that the rack and the pan are not of the same piece. However, in the Notational trains, A/178/5 and A/178/4, the small pan is consistently identified as ^{4}**K** which indicates that the rack and small pan are indeed of a piece. Despite these and other apparent anomalies the clear indication is that the small pan does, in some portrayals, enjoy three degrees of linear freedom in the horizontal plane.

The upshot of these considerations was a decision to split the carriage into two with one travelling platform for the small pan, and one for the large. Both rack and pinion drives remain permanently engaged (Fig. 6.5), and simultaneous stereotyping using both large and small punch wheels becomes possible.

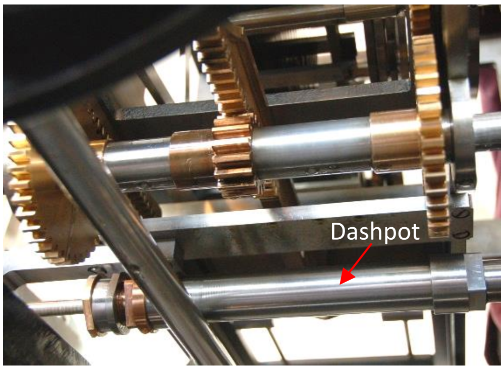


Fig. 6.5: Split carriage drive (underside).

Because of the differing rates of feed of the two pans the large pan gains on the small pan as stereotyping progresses. Calculation confirms that the pans do not crash. A/147 End View shows the distance between the first lines of the small and large pans as 17¼". Working back from the diameter of the pinions gives the maximum travel of the large pan as 14.75", and 5.9" 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 clearance between the pans at the end-of-page is 0.1375" (337 L 25) i.e. the large pan stops short of fouling the small pan. Also, because of the compressed rate of feed of the small pan, the punch wheels do not foul the trailing lip of the pan.

The absence of detail of the fixings of the pans (large and small) could be a simple omission, or could indicate that no fixings are needed with the large pan being driven line-to-line by the front lip of the frame, and the small pan by the central rack and pinion (itself carried with the travelling platform) with the small pan free to slide within the carriage frame.

The general arrangement of the new carriage is shown in General Assembly drawing 337 L 25. Detail of the small pan rack is given in 337 L 352 (called 'rear' i.e. away from the front of the machine), and that of the large pan racks in 337 L 351 A(LH) and L 351 B(RH). As drawn in A/147 the rear (small) pan is some distance away from the rack that drives it and the gap needs to be bridged. The rack is fixed to a ski-shaped support ribbed for stiffness (Fig. 6.3) (General Assembly A 337 L 25) in view of the long overhang. The top surface of the bronze rack-support slides on the underside of the front carriage which acts as a guide. In the split carriage arrangement, the front and rear carriages are of equal size i.e. both carriages can accommodate the large matrix pan. Spacers are used to mount the smaller pan in the carriage frame.

The sleeve ${}^{5}\boldsymbol{B}$ shown as continuous on shaft ${}^{7}\boldsymbol{A}$ (A/147 Plan) was split in 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 (Fig. 6.5). The full-length key was 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 axial drag as the carriage traverses column-to-column.

Column-to-column drive for the whole carriage (right to left in A/147 Elevation and A/163) is less problematic. The carriage is driven by rack ${}^{2}\mathbf{U}_{2}$ and pinion ${}^{1}\mathscr{K}$ at the front of the apparatus, and rack ${}^{2}\mathbf{U}_{1}$ and pinion ${}^{1}\mathscr{N}$ at the rear. The racks and pinions operate in tandem with both pinions driven by the same shaft, ${}^{1}\mathscr{K}$ (Fig. 6.6).

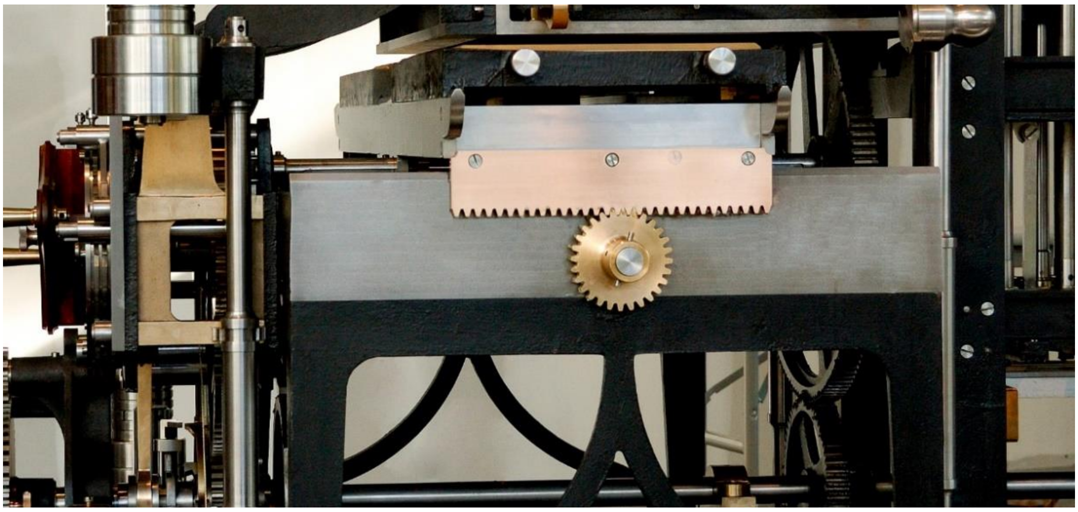


Fig. 6.6: Column-to-column rack and pinion, Elevation.

6.2 Drive and Control

The control mechanism for the stereotyping table is shown in the three views in A/166. The mechanism is driven by the action of two horizontal bars, $\mathscr{E}$ and[9] $\mathscr{F}$, also called oscillating bars (A/166 End View (Fig. 6.8), Figs. 6.7, 6.9) and A/166 Plan (Fig. 6.12). The oscillating bars and the cluster of levers and trips shown between the two pattern wheels (A/166 End View) orchestrate the behaviours of the travelling platform for the various page formats including end-of-page, end-of-line, and end-of-column actions, automatic rewind of the hanging weights, and the normal progressions of the platform during tabulation.

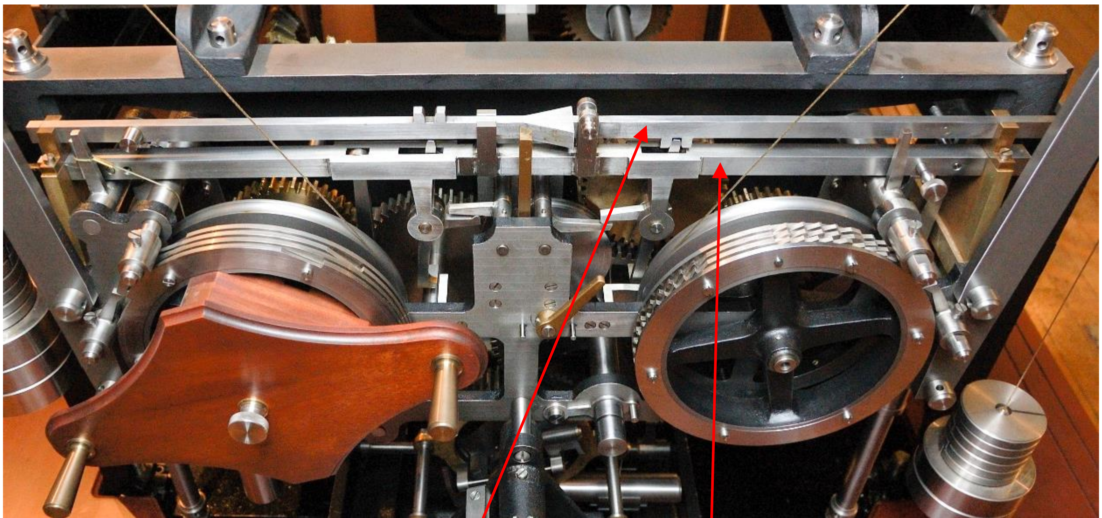


Fig. 6.7: Oscillating bars (oblique).

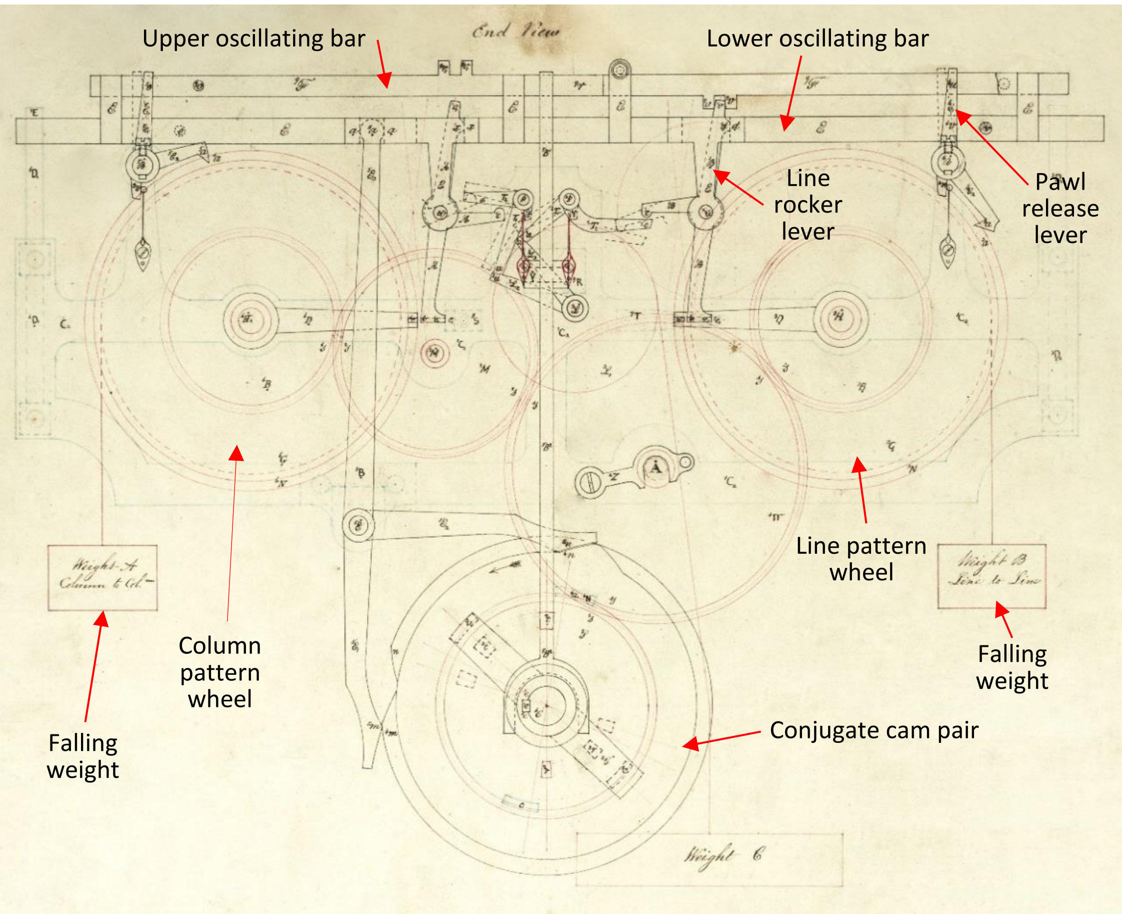


Fig. 6.8: Travelling platform control, (A/166, End View).

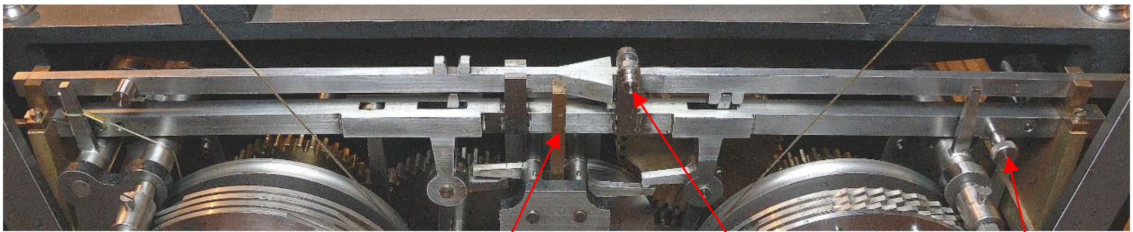


Fig. 6.9: Oscillating bars.

The upper bar ($^{9}\mathscr{F}$) is supported by four guides of a piece with the lower bar and the two bars can slide in relation to each other. The lower bar, $\mathscr{E}$, executes intermittent linear reciprocating motion i.e. moves to and fro' (left-right-left in A/166 End View) all the time the main engine crank is operated. The lower bar $\mathscr{E}$ is driven by lever $^{5}\mathscr{E}_3$ turning on fixed rocker shaft $^{5}\mathscr{E}$. The wishbone cam followers $^{5}\mathscr{E}_1$, $^{5}\mathscr{E}_2$ are driven by a pair of conjugate cams, $^{4}\mathscr{R}_2$, $^{4}\mathscr{R}_1$ (A/166 Elevation, Fig. 6.17). The cam-followers, and rocker shaft, convert the rotary motion of the cams into reciprocating motion of the lever $^{5}\mathscr{E}_3$ (A/166 End View).

The rounded head of lever $^{5}\mathscr{E}_{3}$ is trapped in a recess in the lower oscillating bar. Since the conjugate cams are driven directly from the main drive shaft, $^{4}\mathscr{C}$**,** on the underside of the engine (A/163), the oscillating-bar drive is active all the time the Engine is running and the bar executes one linear reciprocating oscillation each calculating cycle.

The pattern wheels ($^{3}\boldsymbol{N}$, $^{6}\boldsymbol{N}$) are shown as large discs (red) immediately below the two oscillating bars (Fig. 6.8). The line-to-line motion and the column-to-column motion of the travelling platform is determined by the arrangement of teeth on the pattern wheels. One tooth on the pattern wheel corresponds to one stepped increment of the associated matrix pan. The right-hand pattern wheel ($^{3}\boldsymbol{N}$) determines line-to-line motion; the left hand pattern wheel ($^{6}\boldsymbol{N}$) determines column-to-column motion. So one tooth on the line-to-line pattern wheel corresponds to a one line advance. Similarly, one tooth on the column-to--column pattern wheel corresponds to a one-column advance. The pattern wheels are driven by two falling weights: $\mathscr{A}$ drives the column-to-column pattern wheel, $\mathscr{B}$ drives line-to-line action. The weights are suspended by chords over pulleys $^{6}\boldsymbol{G}$ and $^{3}\boldsymbol{G}$ fixed to the pattern wheel shafts ($^{6}\boldsymbol{H}$, $^{3}\boldsymbol{H}$) (A/166 Plan and Elevation).

The oscillating bars control the release of the pattern wheels to advance tooth by tooth under the pull of the falling weights. In the case of line-to-line control, in each oscillating cycle, a peg (Fig. 6.9 right) on the lower oscillating bar releases the line catch pawl, $^{6}\mathscr{C}_{2}$, by bearing on release lever $^{6}\mathscr{C}_{1}$ when it moves right to left (A/166 End View). This frees the pattern wheel to rotate clockwise driven by the falling weight.

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 (the lug is shown in A/166 End View in solid outline, and as a dotted square just below notation $^{6}\mathscr{B}$ in A/166 Plan bottom right). The motion of the pattern wheel is transmitted to the carriage drive shaft $^{7}\boldsymbol{A}$ (A/147) via a gear train. One gear ($^{3}\boldsymbol{B}$) is mounted on the pattern wheel shaft $^{3}\boldsymbol{H}$ (A/166 Plan View). The second gear is shown in (red) outline only with no alphabetical identifier, drawn approximately concentric with the rocker shaft for lever $^{1}\mathscr{B}$ (A/166 End View). This gear, labelled $^{7}\boldsymbol{U}$ in the notation of trains in BAB/A/178/5), drives the travelling platform drive shaft $^{7}\boldsymbol{A}$ i.e. is the last element in the train connecting the pattern wheel to the line-to-line motion of the pans.


Fig. 6.10: Leaf spring.

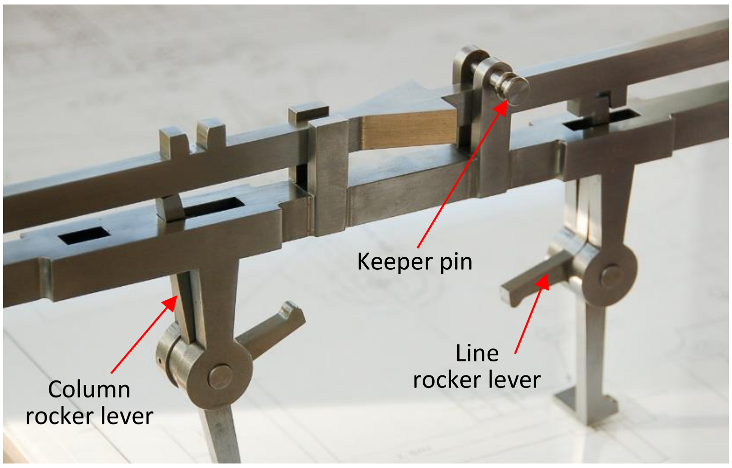


Fig. 6.11: Line and column rocker levers.

For line-to-line tabulation, stereotyping progresses down the page with successive releases of the line pattern wheel. The lower oscillating bar ($\mathscr{E}$) moves to and fro once each cycle and carries with it the line rocker lever, $^{1}\mathscr{B}$ (A/166 End View), the pivot of which is integral with the lower bar and so moves with it. The lower bar has a short section widened to accommodate a through-slot (A/166 Plan) and the line rocker lever passes through the bar to engage a slot in the upper bar (working points $^{9}\boldsymbol{v}$, $^{1}\boldsymbol{v}$, A/166 End View, Fig. 6.11).

For tabulation down the page, uninterrupted by end-of-column or end-of-page conditions, the outward stroke (right to left in A/166 End View) of the lower oscillating bar ($\mathscr{E}$) drives the pad of the line rocker lever ($^{1}\mathscr{B}$) against the right-hand wall of the through-slot (working points $\boldsymbol{d}$, $^{1}\boldsymbol{d}$). The rocker lever is trapped at the angle drawn (A/166 Plan) and the lower bar drives the upper bar, via the rocker lever, to follow the outward stroke (right to left). The upper and lower bars and rocker assembly move as a whole 13/16" to the left. On the return stroke the lower bar moves to the right. The line rocker lever is held by the upper bar and briefly turns anti-clockwise on the moving pivot leaving the upper bar stationary. This continues until the foot of the line rocker lever meets a fixed end-stop which prevents the foot travelling further to the right. The obstruction of the foot by the end stop is almost immediate. (The end-stop is shown as a zigzag series of right angles drawn in outline in faint blue, A/166 Plan, colour added in Fig, 6.12). With the foot of the line rocker obstructed, the line rocker lever drives the upper bar to the right and the line rocker returns to the inclined position shown. The overall effect is that the two bars execute the outward and return stroke together with the upper bar slightly lagging the lower bar at the start of the return stroke. The length of the stroke is the same (13/16") for both bars.

Runaway

When the line catch releases the pattern wheel there is a danger of the pattern wheel running away under the pull of the hanging weight. As drawn (A/166 End View, Fig. 6.8), 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 given that the Engine is manually driven and the speeds inevitably vary. There is also the risk of runaway if the machine is halted with the line catch withdrawn. The column-to-column catch, ${}^{5}\mathscr{C}_{2}$, runs the same risk.

In the Timing Diagram (F/385/1) the line-to-line catch (${}^{6}\mathscr{C}_{2}$) and the column-to-column catch (${}^{5}\mathscr{C}_{2}$) are called 'Detents' and this appears as a column heading for Column 24. The function of the catches is to allow the pattern wheels to advance one tooth, and only one tooth, each cycle in one direction, and to be overridden, against the action of the leaf springs, in the other direction when the weights and travelling platform are rewound back to start-of-page. Their function is therefore somewhere between an escapement and a ratchet. Calling the catches 'detents' might ordinarily imply that they can be overridden in either direction and this is perhaps misleading. In the form as drawn, the ratchet action is there, but not the action of an escapement.

Given the lengths gone to elsewhere (locking mechanisms, for example) to secure reliability, it was not thought prudent 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. The absence of a means of ensuring single-increment advance of the pattern wheel was taken as a design oversight inviting remedy.

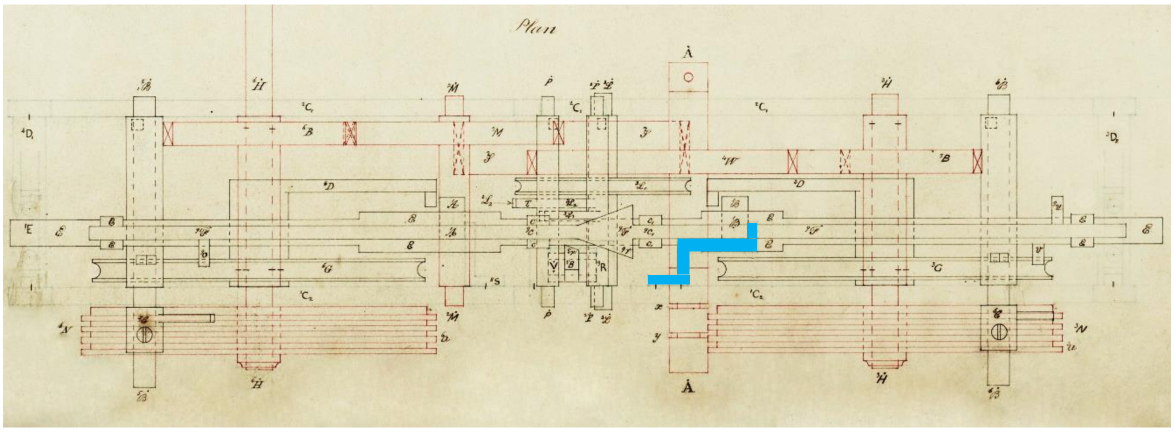


Fig. 6.12: Travelling platform control, end-stop (A/166, Plan) (blue colour added).

Line-to-Line Runaway Pawl

An additional pawl and leaf spring were fitted to provide escapement action i.e. to restrict the pattern wheel advance to one tooth increment only for each release of the catch. The existing line-to-line catch was retained in its original unmodified form and a second line catch (here called the runaway pawl) fitted to prevent unwanted advance after each release (Fig. 6.13). The modified arrangement is shown on General Assembly 337 L 24. The pattern wheel and the two catches act as a ratchet and double pawl. An additional peg in the lower oscillating bar is provided to operate the runaway pawl catch.

In the wait state (i.e. while the travelling platform is stationary while an impression is being taken) the runaway pawl is held out of engagement, against the action of its leaf spring, by its peg in the lower oscillating bar. On the outward stroke (right to left) the runaway pawl is released into engagement on the action of its leaf spring and both the pawls now bear on the same pattern wheel tooth. The continuing outward stroke then releases the line catch pawl ($^{6}\mathscr{C}_2$, working points 𝓾 and ^{6}𝓾 A/166 End View) acted on by *its* peg to allow the pattern wheel to advance. During the pattern wheel advance the runaway pawl acts as a security backstop. The return stroke (left to right) follows immediately. This releases the line catch pawl to engage under the action of its leaf spring and the continuing return stroke then takes the runaway pawl out of engagement. The runaway pawl engages before the line catch pawl is withdrawn (outward stroke) and disengages after the line catch pawl has re-engaged (return stroke). Both pawls must act in close, lapped, succession, on the same pattern wheel tooth for each release, to ensure that the backstop action is effective for the smallest tooth pitch.

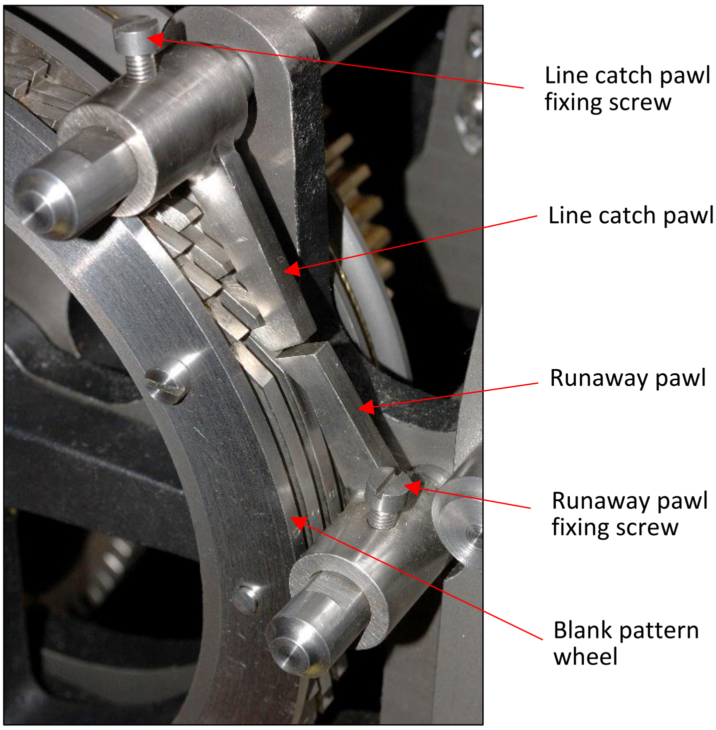


Fig. 6.13: Line-to-line runaway pawl.

The guides for the oscillating bars require modification to accommodate the additional peg. The lower bar is shown supported at each end by channels in the two end supports (Fig. 6.9, A/166 End View), and the upper bar supported by four sets of guides ($\mathscr{E}$) attached to the lower bar (A/166 Plan). To allow space for the extra peg in the lower bar, the outer two sets of guides are combined with the end supports to provide support for the upper and lower bars (337 L 471, L 472); the central two remaining guides remain unchanged.

Column-to-column Runaway Pawl

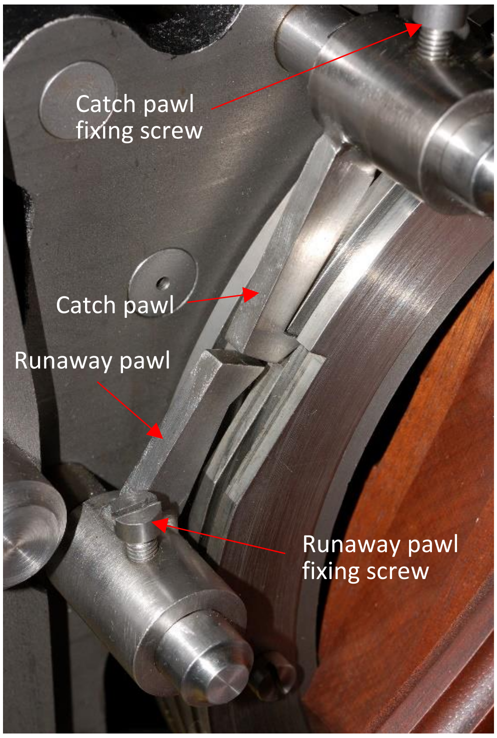


Fig. 6.14: Column-to-column runaway pawl.

The modification of the column-to-column catch is more extreme though the principle of providing an additional pawl is identical. The pattern-wheel ($^{6}\boldsymbol{N}$) assembly and original catch ($^{6}\mathscr{C}_1$) is more closely crowded towards the end framing pieces than the line pattern-wheel assembly (A/166 End View) and there is insufficient space for a second column catch without modification to the original catch.

The modified arrangement is shown in 337 L 24 (General Assembly) and specific layouts of pawls and levers in 337 X 24 and X 25). The modified catch is reversed i.e. is changed from a leading action that blocks the pattern wheel advance, to trailing edge action holding back the pattern wheel. The column catch is also driven differently: the original catch ($^{6}\mathscr{C}_1$) is operated by a peg in the lower oscillating bar (working points $\boldsymbol{e}$ and $^{5}\boldsymbol{e}$ A/166 End View top left) i.e. by direct lever action. The direct lever action has been changed to a gear-driven lever with parts shown in 337 L 491-493. An additional runaway pawl is provided (337 L 24) that has a leading-edge action i.e. blocks pattern wheel advance when engaged. The second catch is also a gear-driven lever (337 L 494-497), itself driven by additional pegs in the lower oscillating bar. The original pivot position of the first catch is unaltered and additional pivots are provided for the pawl lever and second catch. The phasing of the two catches is identical to that of the line catches i.e. the runaway pawl engages before the first catch is withdrawn and is withdrawn immediately the first catch re-engages. The conversion of the first column catch to trailing action has the incidental elegance of restoring the symmetry with the modified line catch arrangement opposite.

The modifications have minor consequences for the pawl profile and pattern wheel teeth. The tooth on the original line catch ($^{6}\mathscr{C}_2$)s shown hooked for ratchet action, and a sawtooth on the reverse side so that the catch can be overridden against the leaf spring during rewind (A/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) to have radial tooth rises on the active faces. The original arrangement of one leading and one trailing lever required different tooth profiles: standardising the pawl profile also means that all the pattern wheels (line and column) can have the same tooth profiles.

Pattern wheel selection is made by unscrewing the fixing screws (Fig. 6.13, A/166) and sliding the two pawls (the original catch pawl and the additional runaway pawl) on shafts $^{6}\boldsymbol{\mathscr{B}}$ (line-to-line) and $^{5}\boldsymbol{\mathscr{B}}$ (column-to-column) to engage with one of the pattern wheels in the pre-mounted sets of four (see **User Manual (2013)**, **Setting Formatting Options**, pp. 49, 51-3).

End-of-column Action

For line-to-line tabulation (i.e. down the page) when a pan reaches the end of a column (bottom of the page) the mechanism engages a rewind clutch that automatically rewinds one of the falling weights and also rewinds the pans back to top of page. The rewind clutch ($^{3}\mathscr{F}$) is shown in A/166 End View and Elevation, and A/163 bottom left.

Integral with the pattern wheel is a line pattern wheel arm ($^{3}\boldsymbol{D}$ A/166 Plan and End View) which rotates with the pattern wheel. Each release of the pattern wheel rotates the arm one increment clockwise, by an amount determined by the pitch of the pattern wheel teeth, and this continues with the pattern wheel arm stepping its way clockwise until it approaches the nine o'clock position. The line rocker lever is displaced to the left when the pattern wheel is next released and a lug on the pattern wheel arm fouls the underside of the foot of the line rocker lever (for foot shape see Fig. 6.11). This holds the pattern wheel arm just off the nine o'clock position. On the return stroke the foot of the rocker arm moves to the right and frees the obstruction caused by the pattern wheel arm which advances to nine o'clock placing the lug ($^{3}\boldsymbol{D}$ A/166 Plan) immediately behind (left) of the foot in the position shown in A/166 End View and Plan. The foot of the rocker lever is now trapped between the fixed end stop (Fig. 6.12) to the right and the lug of the pattern wheel arm to the left. 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 (left) stroke of the lower bar throws the upper bar left beyond the 13/16" by the lever action of the line rocker lever turning on the moving pivot.

In the case of multiple-column stereotyping the extended throw of the upper bar at the end-of-column, releases the column pattern wheel to rotate, driven by the falling weight $\mathscr{A}$ **(**A/166 End View) one tooth anti-clockwise. The catch is released by a fixed pin on the upper oscillating bar acting on the catch release lever (working points $^{5}\boldsymbol{a}$, $^{9}\boldsymbol{a}$ A/166 End View top left). This drives the travelling platform to the next column along (right to left in A/147 Elevation) i.e. drives the carriage across the page, right to left as seen from the front of the Engine, to position the matrix pan in the new column position.

As in the case of the line-to-line pattern wheel there is no provision in the original design to prevent runaway in the event of incorrect engine speed or the engine cycle halting with the column catch disengaged, and a second column catch was added to provide an escapement action as a precaution to prevent runaway (see **Runaway**, p. 113). The column-to-column drive from the pattern wheel to the carriage is via bevel gears (***F*** A/163) and in outline (red) in A/147 End View). The column pattern wheel directly drives the shaft $^{6}\boldsymbol{H}$ (A/166 plan) which drives the cross-shaft $^{1}\mathscr{K}$ (A/163, A/147 End View) via the bevel gears. The column-to-column drive train is completed by the racks and pinions at each end of the frame and shown in red (Fig. 6.5, A/147 End View).

Rewind Clutch

The second effect of the extended throw of the upper oscillating bar is to engage the rewind clutch $^{3}\mathscr{F}$ A/166 End View and Elevation, and A/163 bottom left. This is a dog clutch that engages to automatically rewind one or another of the falling weights and, via the pattern wheels, rewinds the travelling platform to the top of page. The clutch is mounted on the main drive shaft $^{4}\mathscr{C}$ that runs on the underside of the engine (A/163 , A/166 Elevation and End View). A clutch bar (Fig. 6.15), which is part of the bobbin $^{3}\mathscr{F}$ (A/166 Elevation and shown diagonally top left to bottom

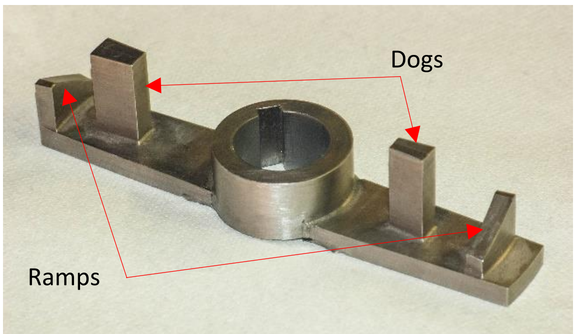


Fig. 6.15: Clutch bar for rewind clutch.

right A/166 End View), is fixed to the main drive shaft ($^{4}\mathscr{C}$) by a sliding key, and rotates continuously during a calculation run. The bobbin and clutch bar are able to slide along the shaft on the sliding key.

The clutch is shown disengaged in A/166 Elevation (Fig. 6.17) i.e. the inner lugs on the clutch bar are clear of lugs on the stationary gear wheel, $^{2}\mathscr{S}$ (A/166 Elevation, Fig. 6.17, in red). The clutch is engaged by the clutch lever ($^{5}\boldsymbol{B}^{2}$, $^{5}\boldsymbol{B}^{1}$ A/166 Elevation and End View, Fig. 6.23) when the forked end of the clutch lever ($^{5}\boldsymbol{B}^{2}$), pivoting on short shaft **V** (A/166 End Elevation and End View) drives the clutch bar so that the inner two lugs on the clutch bar are in the same plane as the matching lugs on the stationary gear wheel (left to right A/166 Elevation). The clutch lever (Fig. 6.23) is operated by the ramp on the upper oscillating bar shown as an arrow shape in A/166 Plan. The lower ramped edge (working points $^{9}\boldsymbol{\mathscr{u}}$, A/166 End View and Plan) throws the top of the lever out when the upper oscillating bar is thrown forward (right to left A/166 End View) by the end-of-column condition. The lever-action on pivot **V** drives the forked lever to slide the clutch bar towards the stationary gear wheel.

As the clutch bar sweeps round, the two inner lugs on the bar engage with two lugs on the stationary gear wheel $^{2}\mathscr{S}$ (A/166 Elevation, in red) (working points $^{3}\boldsymbol{\mathscr{t}}$, $^{2}\boldsymbol{\mathscr{t}}$ and $^{3}\boldsymbol{\mathscr{u}}$, $^{2}\boldsymbol{\mathscr{u}}$ A/166). The lugs are 180° displaced and on different pitch circle radii so that the outer and inner lugs only engage with each other. The clutch bar drives the gear wheel $^{2}\mathscr{S}$ anti-clockwise (A/166 End View) which rewinds the line pattern wheel via gear $^{4}\boldsymbol{W}$. Only one pattern wheel is automatically rewound: $^{4}\boldsymbol{W}$ 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 $^{4}\boldsymbol{W}$ along its shaft and dropping the change-over latch $^{2}\boldsymbol{Z}$ into one of two slots in the spacing sleeve on fixed shaft $\mathbf{A}$ (A/166 End View, Fig. 6.16). The extra thickness of gearwheel $^{2}\mathscr{S}$ ensures that $^{4}\boldsymbol{W}$ 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-

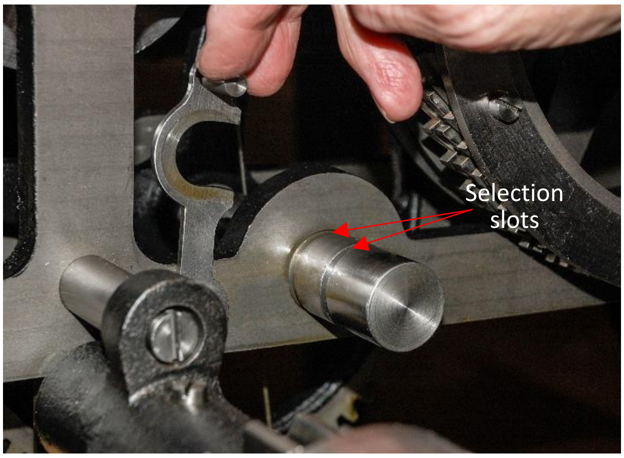


Fig. 6.16: Change-over drop-latch for rewind clutch.

column stereotyping, the column pattern wheel is rewound automatically and the line-to-line pattern is rewound by hand at the end-of-page. (For detailed step-wise procedures see **User Manual (2013), Converting from line-to-line to column-to-column**, p. 58).

The two actions that follow from the throwing forward of the top oscillating bar when the end-of-column condition occurs are: releasing the column pattern wheel to position the pan to receive the last line of the column; and engaging the rewind clutch that rewinds the line-to-line pattern wheel so positioning the matrix pan to receive the next stereotyped result as the first entry of the new column at top of page.

The clutch is thrown out of engagement after about 340° rotation by two fixed ramps (^{6}**N** blue pencil A/166 Elevation and End View, Figs. 6.17, 6.8) engaging with mating ramps on the clutch bar (working points $^{3}\boldsymbol{a}$, $^{6}\boldsymbol{a}$ and $^{3}\boldsymbol{c}$, $^{6}\boldsymbol{c}$ A/166 Elevation). 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 by the action of the two ramps, ^{6}**N**. The line pattern wheel is held in the rewound position, with the weight raised, by the first 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 driving the matrix pans from top of page to bottom so that when the pattern wheel arm is at the start position i.e. when the pan is at top of page, the arm is past (clockwise) the nine o'clock position: there are no spurious end-of-column actions during a page run as the pattern wheel arm only reaches nine o'clock once in a traverse down the page. 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 wheel arm rather than the pattern wheel is the effective component. Once set at three o'clock at the start of page, the column pattern wheel arm ($^{6}\boldsymbol{D}$, A/166 End View) remains in this position to trigger the end-of-page

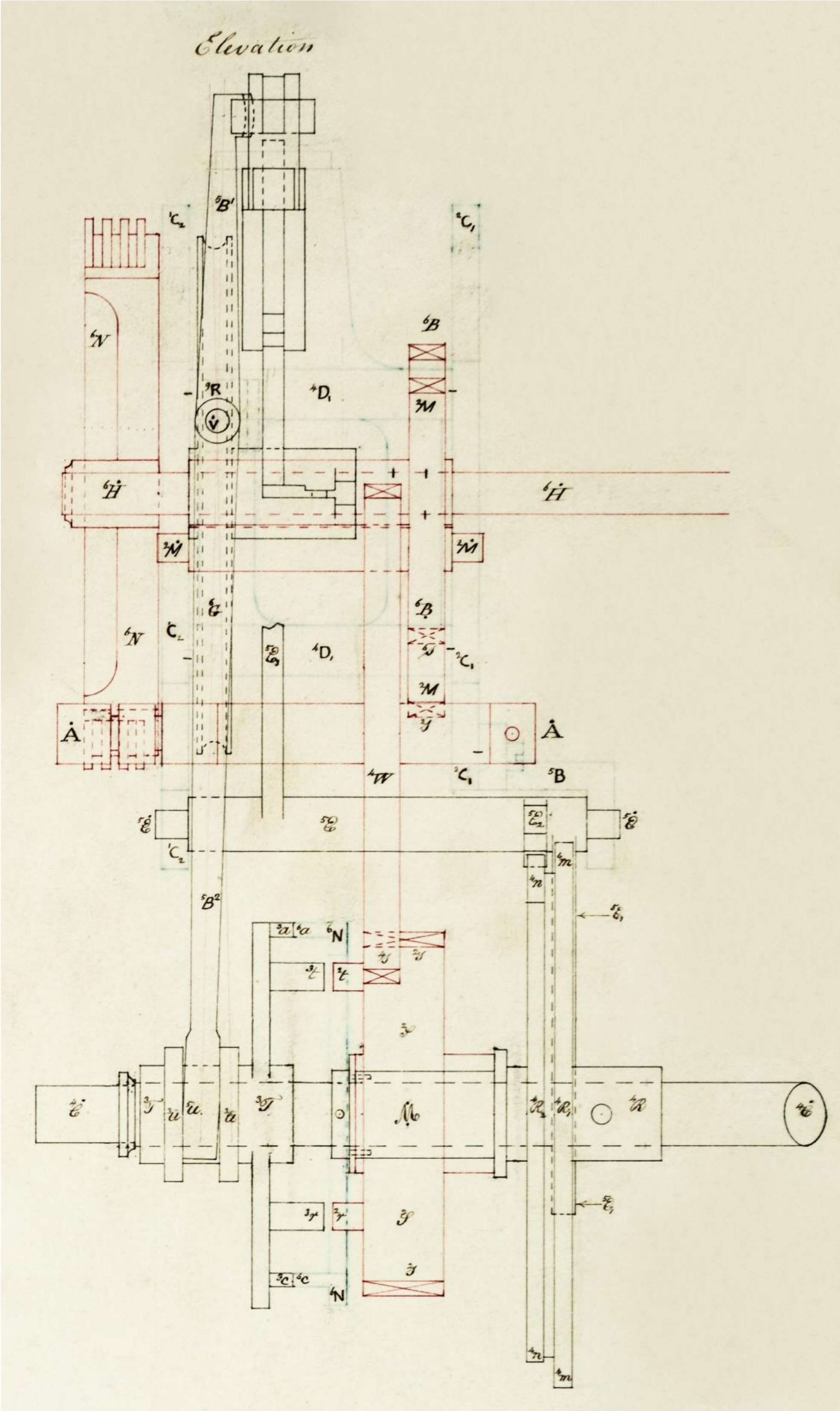


Fig. 6.17: Travelling platform control, Elevation (A/166).

sequence. For column- to-column stereotyping, the column pattern wheel is rewound automatically at the end of each line and 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.

Modifications to the Rewind Clutch

In the original design the clutch-driven rewind returns the carriage from the end-of-page position to top-of-page 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, thirty-line formats pose difficulties. Only about half the circumference of the thirty-line pattern wheel will have teeth (194° from first to last tooth, 337 L 409) with the remaining sector blank. Since a run always finishes with the carriage at the limits of its travel (from front-to-back 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 engages – something to 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 consequences of fixed rewind still affects operation in column-to-column mode. Once again, the last column position is the fixed reference and it is the start positions for two and three column formats that 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 the column-to-column traverse of the carriage.

Variable Rewind

If the amount of rewind could be made adjustable then the carriage could be rewound by the clutch to an appropriate start position and both the potentially damaging situations avoided. An additional advantage of an adjustable partial 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 column pattern wheel does not move.) The column pattern wheels can then be used to provide a wider range of margin-width options.

The time window for any rewind action is fixed by the engine speed i.e. the speed of the main drive shaft $^{4}\mathscr{C}$. How much rewind takes place within the window depends on the start and end times of the rewind action. Rewind action starts when the lugs on the rotating clutch bar encounter the rewind lugs on the gear wheel $^{2}\mathscr{S}$; rewind action stops when the clutch is disengaged i.e. thrown out of engagement by the two ramp lugs acting on the outer lugs of the clutch bar. Reducing the amount of rewind can be achieved by delaying the start of the rewind action, or advancing its end point. The first can be achieved by making provision for adjusting the angular position of the clutch bar in relation to the main drive shaft $^{4}\mathscr{C}$. Adjusting the clutch bar anticlockwise (A/166 End View) will delay the start of the rewind action keeping the end point fixed. Advancing the end point can be achieved by making provision for adjusting the angular position of the fixed ramp lugs (^{6}**N** blue pencil A/166 End View). Adjusting these clockwise will advance the disengagement of the clutch and so reduce the amount of rewind.

There is no conclusive evidence from the drawings as to whether the original design was intended to cater for variable rewind. A/166 End View and Elevation show two rewind lugs (working points $^{2}\boldsymbol{t}$ and $^{2}\boldsymbol{u}$) but these are shown integral with the gear wheel $^{2}\mathscr{S}$ with no provision for adjustment. Similarly there is no evidence of provision for adjusting the angular position of the clutch bar in relation to the main shaft $^{4}\mathscr{C}$. There is the suggestion of a backplate supporting the two ramps that disengage the clutch (^{6}**N** blue pencil A/166 End View) which could be a placeholder for a detail to follow of how the backplate may be fixed in different positions. But no detail is provided.

The upshot was to go ahead with provision for variable partial rewind 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 four column pattern-wheels. The means of doing so was to make provision for adjusting the angular position of the rewind lugs on the gear wheel $^{2}\mathscr{S}$(Parts details 337 L 521 (M 383), 337 L 522). Instead of the rewind lugs being integral with the gear wheel, the lugs are mounted on a backplate the angular position of which can be varied in relation to the gear.

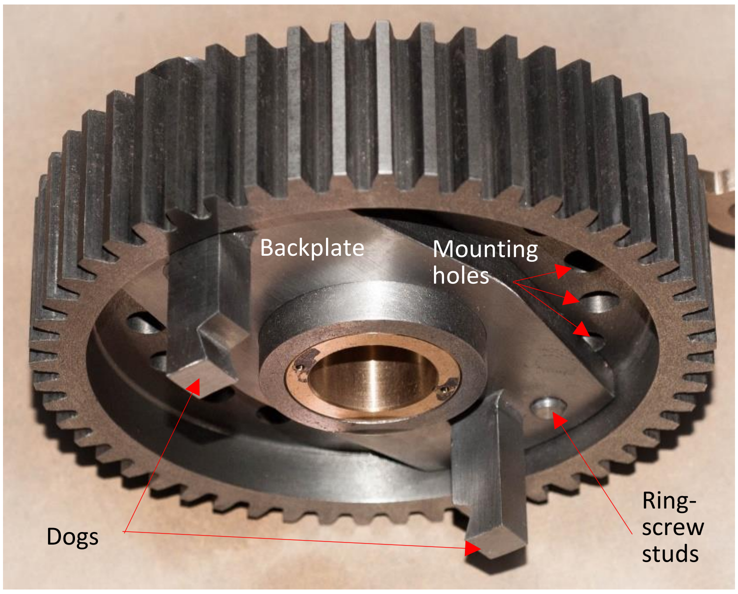


Fig. 6.18: Modified clutch gear.

The web of the clutch gear is drilled with eighteen holes at 20° intervals, and the backplate, which nests inside the gear, is fixed to the web by two ring screw studs which pass through any of the pairs of diametrically opposite holes (parts details 337 L 521 and L522). To adjust the amount of rewind the plate is rotated until the engraved mark lines up with the pattern wheel letter on the clutch gear. The ring screw studs fix the plate in the new position. The eighteen holes provide a series of fixed 20° angular increments through which to reduce the amount of rewind. The further clockwise the backplate is fixed the shorter the rewind.

Redesign of the Rewind Clutch

The clutch bar is intended to slide into engagement without load, drive the clutch gear via two lugs, and then disengage under load driven by ramps. In trials it was found that under load, sliding friction between the faces of the engaged lugs, and bobbin friction on the keyway and shaft prevented the clutch bar from disengaging i.e. under load the angle of the ramps was too steep to exert sufficient force to slide the clutch bar and bobbin out of engagement. Finger-tip rotational pressure was sufficient to produce a jam indicating that keyway friction was almost certainly the main cause of the problem.

The clutch was redesigned to reduce the force required to disengage by:

1. Eliminating keyway friction: the bobbin is no longer keyed to the shaft
2. Reducing the contact area between the driving and driven lugs
3. Using shallower ramps i.e. smaller lead angle, and helical profile.

The new mechanism is shown in Figs. 6.19, 6.20. The bobbin is no longer keyed to the drive shaft but fixed, and the clutch bar (Fig. 6.15) is replaced by a pawl assembly consisting of an inner pawl (337 M 385) and an outer pawl to provide engagement with the two clutch gear lugs which have differing pitch circles as in the original design. Integral with each pawl is one half of the ramp. The pawls rotate separately on spigots in the pawl housing (M386). The clutch gear remains unchanged but the dogs in the modified clutch are clawed.

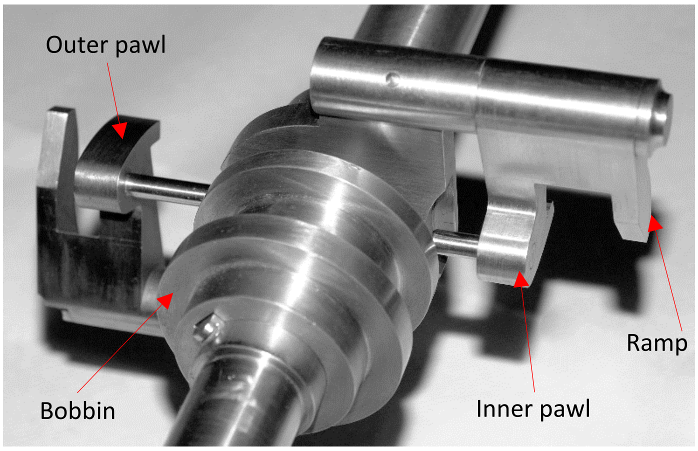


Fig. 6.19: Modified bobbin and pawls.

The pawl assembly is keyed to the shaft and rotates continuously. Each pawl has a pin on the centre-line through the large drive shaft, which projects into the axial path of the bobbin flange. To engage the clutch the bobbin slides towards the clutch pushed by the forked yolk. The bobbin flange pushes the pawl pins to rotate the two pawl claws into the path of the lugs fixed to the clutch gear. This is done ahead of engagement so the load on the forked arm is small as it is only required to turn the unloaded pawls a few degrees. As the pawl assembly rotates the pawls engage with the rewind lugs and the clutch gear is driven anticlockwise. The pawls stay engaged until thrown out of engagement by the stationary fixed ramps on the ramp ring (337 L 512).

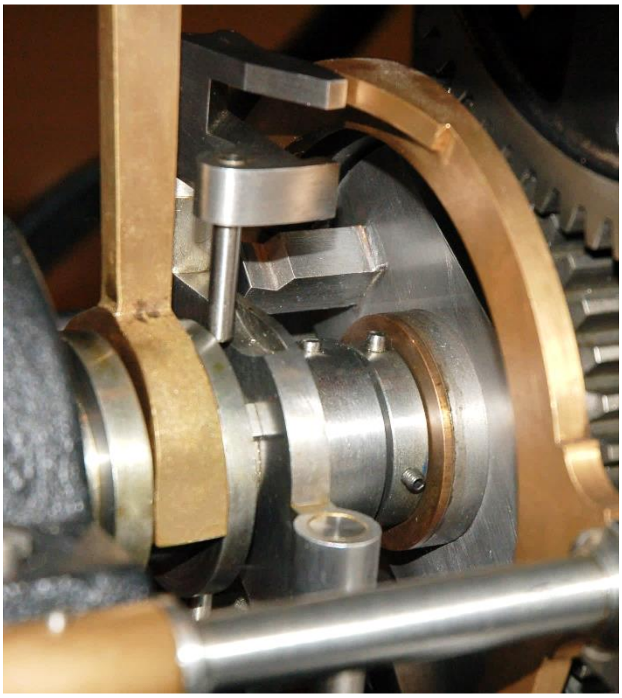


Fig. 6.20: Modified rewind clutch assembly.

The lead angle on the ramps was reduced from 11.5° to 7.5° which produces less axial travel to throw the pawls out than in the original design as less travel is needed to disengage the dogs than previously, and the smaller angle gives a greater mechanical advantage. The contact area between the claw of the pawl and the driven lug is also reduced which reduces the force required to throw the clutch out of engagement.

The bobbin acts on the pawls at 307° into the cycle. For maximum rewind the pawls engage with the driven lugs at 323° i.e. there is a minimum of 16° during which the pawl assembly rotates without load. For shorter rewind options the unused motion is proportionately greater. The lugs take the clutch gear 8° past its starting catch. When the pawls disengage, the clutch gear drops back 8° and the weight is held raised.

The rewind clutch was the only device in the original design that required modification through redesign. The new mechanism fits into the same space, is based on the same principle, and was devised to ensure reliable disengagement.

End-of-page

With line-to-line operation in multiple columns, 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 above (see **End-of-column Action**, p. 117. If the page is complete at the end-of-column then the control mechanism automatically halts the machine to allow the pans to be replaced.

The end-of-column condition is indicated by the line pattern wheel arm ($^3\boldsymbol{D}$, A/166 End View) reaching the nine o'clock position which initiates the release of the column catch by a peg on the upper oscillating bar operating the column catch release-lever. In the case of multiple-column stereotyping the column pattern wheel increments to the next tooth position and the column pattern-wheel arm ($^6\boldsymbol{D}$, A/166 End View) rotates anticlockwise towards the three-o'clock position; in the case of single column stereotyping the column pattern wheel arm is set at three o'clock at the start of page and remains stationary throughout. The end-of-*page* condition occurs when the line pattern-wheel arm is at nine o'clock signalling end-of-column *and* the column pattern-wheel 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 trips shown between the two pattern wheels (A/166 end view).

Stop Weight

The end-of-page action triggers the release of weight $\mathscr{C}$ (A/166 End View, bottom centre) suspended by a cord over stop pulley $^3\mathscr{L}_1$ (A/166 End View red outline). The engine is halted when the weight is released into the trough (***B***) below (A/163 bottom left, 337 L 27. The falling weight operates the trip lever (***B***) (Fig. 6.21) which, via a chord and pulley arrangement, primes a scoop cam to disengage the main drive clutch at the top of the cam stack. The clutch breaks the drive between the operator's handle and the bevel gear of the cam drive-shaft and the Engine halts with the crank handle turning freely with the Engine stationary in the halted state (see **7.2 Uncoupling Clutch: Automatic Halting**, p. 153). The geometry of the stop weight is uncertain. The depiction of the weights in A/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 A/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 gutter when manually rewinding it and that in its rest position it would dangle clear of the sides making the chord liable to twist under its own weight and fail to re-engage when next lowered. So it was assumed that the weight was intended to be a sphere and the trip lever a form of scoop. It transpires that in its rest position of the stop weight remains contained by the trough as drawn, so a modified cylindrical geometry was chosen (Fig. 6.21).

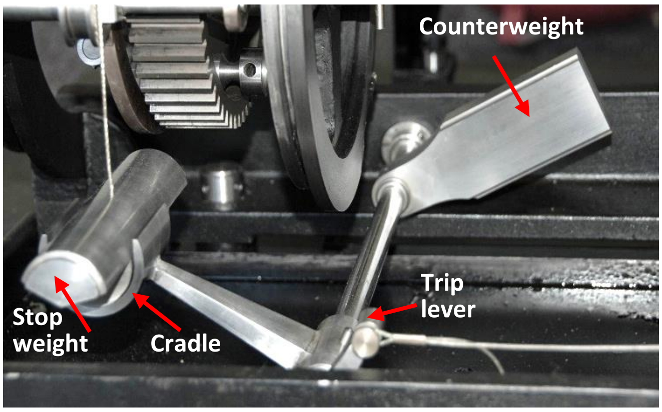


Fig. 6.21: Stop weight trip lever.

End-of-page Control

The stop pulley, ${}^3\mathscr{L}_1$, is controlled by two radial levers (${}^3\mathscr{L}_2$, ${}^3\mathscr{L}_3$ Fig. 6.22, A/166 End View and Plan) both of which are integral with the boss of the stop pulley. The levers are shown between nine and ten o'clock and between ten and eleven o'clock (A/166 End View, Fig. 6.22). One lever (${}^3\mathscr{L}_2$) is the column stop, the other (${}^3\mathscr{L}_3$) is the line stop. Each of the levers is obstructed from clockwise motion by a trip lever ($\boldsymbol{T}_1$, ${}^2\boldsymbol{T}_1$) which rock on fixed pivots ($\boldsymbol{P}$, ${}^2\boldsymbol{P}$) and are each in the same plane as the two related levers. 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 (${}^2\boldsymbol{T}_2$, $\boldsymbol{T}_2$) are operated by extra arms ${}^1\mathscr{B}$, ${}^1\mathscr{A}$ (Fig. 6.22, A/166 End View), integral with the line and column rockers respectively. The arm on the line rocker is shown at roughly nine o'clock (A/166 End View) and that on the column rocker at three o'clock.

The end-of-column condition occurs when the line pattern-wheel arm (${}^3\boldsymbol{D}$ A/166 End View) reaches nine o'clock; the end-of-page condition occurs when the line pattern-wheel arm

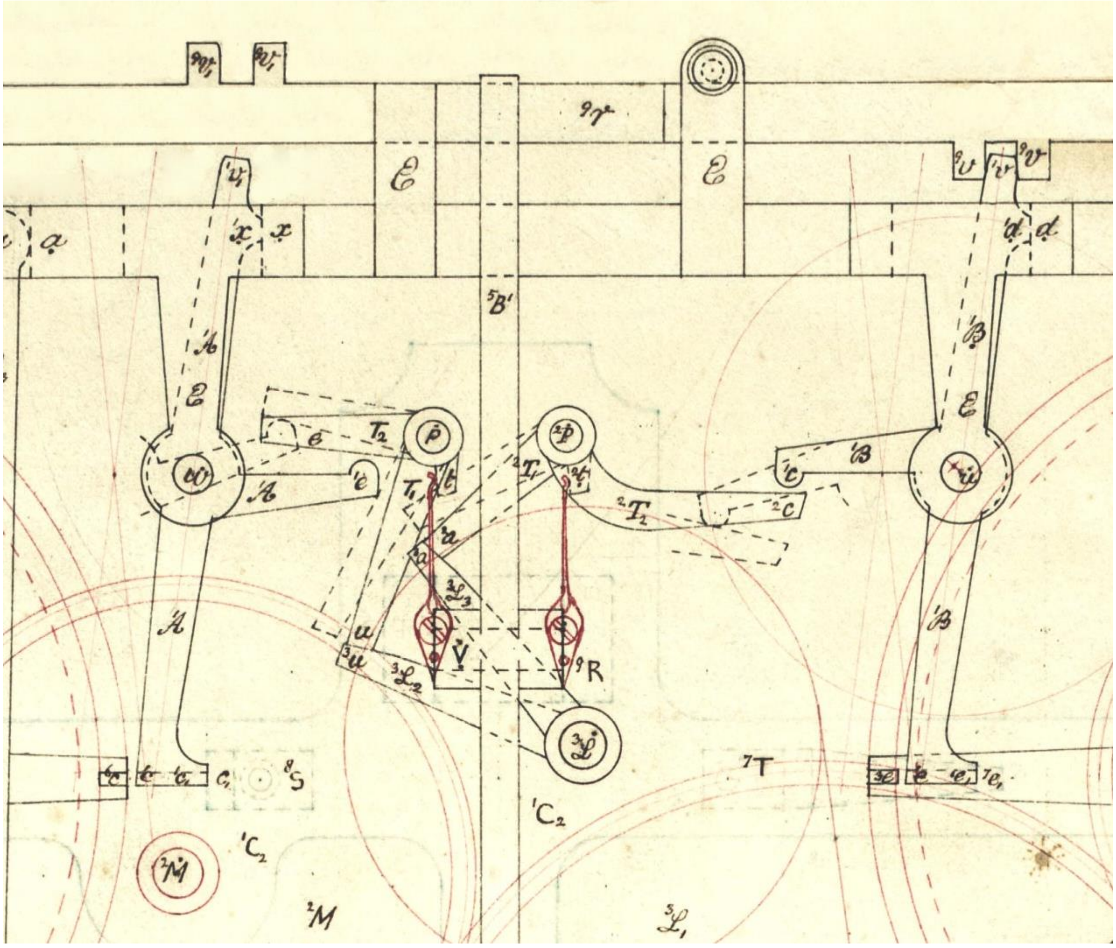


Fig. 6.22: Stop weight release control (A/166, End View) (detail).

reaches nine o'clock *and* the column pattern-wheel 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-wheel 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 (dotted in A/166 End View) operates lever $^2\boldsymbol{T}_1$, $^2\boldsymbol{T}_2$ which releases line stop $^3\mathscr{L}_3$. Similarly, if the column pattern-wheel arm is at three o'clock the column rocker operates the column trip to release the column stop $^3\mathscr{L}_2$. Only if both these conditions are satisfied is the stop pulley ($^3\mathscr{L}_1$) released. If either pattern wheel 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 (right to left) together, the upper bar lags the lower bar during the return stroke. During the hysteresis interval the line rocker (in the line-to-line configuration shown in A/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 the column trip is operated each time the lower bar executes the return stroke again with no net effect.

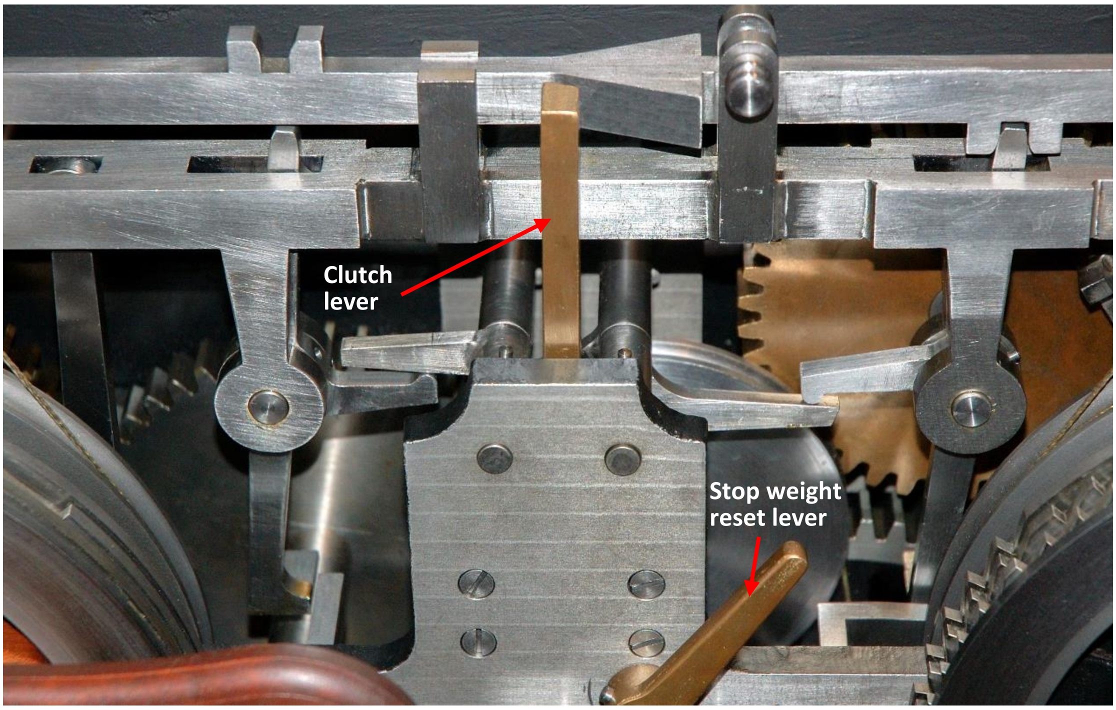


Fig. 6.23: Stop weight release control.

Conversion for column-to-column Stereotyping

The configuration of the mechanism shown in A/166 is for line-to-line stereotyping in single or multiple columns. In this format each result is impressed below its predecessor to generate a column of results. In the case of single column operation, at the end of a full column, rewind to the top-of-page occurs to restart a new page or, in the case of multiple column operation, to start a new column at the top of the same page. 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 (see **2. Overview**, Figs 2.8, 2.9, p. 17).

The conversion from line-to-line to column-to-column operation is made by removing the upper of the two oscillating bars ($^{9}\mathscr{F}$) (Figs. 6.7, 6.8, 6.9), A/166 End View) and refitting it in an inverted position.

The upper bar is freed by removing the keeper pin (Figs. 6.9, 6.11) fitted across one pair of guides shown second from the right in A/166 End View. Once the bar is inverted the keeper is replaced. The two lugs on the upper bar (working points $^{9}\boldsymbol{\mathcal{U}}$ A/166 End View) which hold captive the upper end of the line rocker $^{1}\boldsymbol{\mathcal{B}}$ in the line-to-line configuration, face upwards in the inverted position, and are inactive. Two corresponding lugs to the left ($^{9}\boldsymbol{\mathcal{U}}_{1}$) on the upper bar become active by trapping the end of the column rocker ($^{1}\mathscr{A}$) instead. The conversion is completed by transferring the removable pin on the lower bar to a corresponding position on the left of the same bar 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 gear, $^{4}\boldsymbol{W}$, is repositioned and fixed with the change-over drop catch (Fig. 6.16) so as to disengage from the line pattern wheel and, instead, drive the column pattern wheel via idler, $^{2}\boldsymbol{M}$ (A/166 End View). (See **User Manual (2013)**, C**onverting Format from Line-to-Line to Column-to-Column**, pp. 58-59 for stepwise procedures for the conversion.)

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 $^{6}\boldsymbol{H}$ (A/166 Plan and End View), bevel gears $^{6}\boldsymbol{L}$, $^{1}\mathscr{L}$ ($^{1}\mathscr{L}$ is on cross-shaft $^{1}\mathscr{K}$ A/147 End View. Also A/163), pinions $^{1}\mathscr{K}$ and $^{1}\mathscr{N}$ (pinion $^{1}\mathscr{K}$ is pinned to the cross-shaft and so has the same identifier, $^{1}\mathscr{K}$) and racks $^{2}\mathbf{U}_{2}$ and $^{2}\mathbf{U}_{1}$. (A/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 stop. 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-wheel arm at nine o'clock) this has no effect.

In the case of two-column stereotyping the column pattern-wheel 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 ($\mathscr{A}$) the column pattern wheel arm ($^{6}\mathbf{D}$ A/166 End View) fouls the underside of 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-wheel 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 of the oscillating bar (right to left) the pattern wheel arm obstructs the foot of the column rocker and the rocker levers the upper bar to the left (A/166) in excess of the standard 13/16" travel. The extended throw of the upper bar allows the fixed pin 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 (A/166 plan) to operate the clutch lever ($^{5}\boldsymbol{B}^{2}$ and $^{5}\boldsymbol{B}^{1}$, Fig. 6.23) and engage the clutch which rewinds the column pattern wheel as described earlier.

The sequence described is repeated with a line increment occurring after alternate column pattern wheel releases. The line pattern wheel advances clockwise until the line pattern-wheel arm reaches nine o'clock. The end-of-page condition is met when both pattern-wheel 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 as before. 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 stepwise procedures see **User Manual (2013)**, **Resetting after End of Page**, p. 56-7). 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 matrix pans are more or less easily removed depending on where in the cycle the Engine stops whether the halt is automatic as at end-of-page, whether deliberate by stopping the manual crank, or through a fault. The two most convenient points are 240° to remove and replace the large pan, and 350° to remove and replace the small pan (337 X 23), and rather than struggling to free the pans from under the type heads, it is best that the Engine is advanced to these positions to change pans if the halt position does not coincide with either of these preferred points in the 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 stereotyping with small type the small pan will be run out to the left (A/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 can be eased by re-engaging the main drive clutch, advancing to the next cycle and halting the machine manually at 230°. This corresponds to the completion of the automatic carriage rewind cycle and coincides with the disengagement of the rewind clutch. In this position the large pan is fully run out to the right (A/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 without disturbing the continuity of printed or stereotyped results.

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 in the travelling frame and to move the pan towards the gap between the type heads (left to right A/147 End View). The pan can then 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 A/147 End View.

The account above is based on a reading of the original drawings and timing diagram before construction. Following trials with the built Engine, modifications were made to allow the output apparatus to be uncoupled from the calculating section and driven independently using an auxiliary crank (Fig. 6.1). Advancing the travelling platform and positioning the pans for unobstructed removal and replacement is more conveniently done using the auxiliary crank after the output apparatus has been uncoupled from the main drive. For the rationale and design of the uncoupling mechanism and auxiliary crank see **7. Release clutch and auxiliary drive for output apparatus**, p. 140).

6.3 Formatting Options

The original drawings give no details of formatting options either as page-layout descriptions or as pattern wheel profiles that would realise them. A/163 and A/166 simply indicate a stack of four line-pattern wheels and four column pattern-wheels mounted on the pattern-wheel shafts with no detail of pattern-wheel variations. The format options were determined from scratch and the pattern wheel tooth layout derived from these.

The known constraints are: the size of the two matrix pans (A/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 (A/147 End View); and the width of the column of print for each of the two type sizes (30 by 1/16" for the small pan; 30 by 1/8" for the large pan). (The number of digits in a result is not variable and is always thirty).

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 has the width to accommodate four columns of type but if moved far enough to the right (A/147 Elevation) for a start position close to the left hand edge of the pan, the large stereotyping heads would foul the left-hand edge of the pan at the start (the span of the small type heads is narrower than that of the large type heads but they are on the same centre line, A/173, 337 X 23). If the large pan were 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 in the original design drawings the following formatting features were taken as discretionary: side margins, margin widths between columns for multiple column formats, top and bottom margins, line height, and the number of lines grouped together and separated by a blank line (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, the line pitch, and top and bottom margins are chosen by selecting one of four line-pattern wheels A, B, C or D; column format combinations are chosen by selecting one of four column pattern wheels E, F, G, or H. Each pattern wheel edge is stamped with its identifying letter (337 X 23).

All four pattern wheels in a set are mounted on the pattern wheel shafts and are conveniently exposed for access. Only one pattern wheel from each set of four is active at any one time. The selection is made by sliding the line catch pawl to engage with the chosen pattern wheel (A/166, 337 L 409) (Figs. 6.13, 6.14) (see **User Manual (2013)**, **Setting Formatting Options**, pp. 49-53). Pattern wheel D is left blank deliberately 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 sexagesimal increments (trigonometric and astronomical tabulation) with the thirty-line format and large type used where large line spacing is required. The full set of pattern wheel combinations and the layouts that result are shown on 337 X 23.

Column formatting is controlled 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 L409 A-D (line pattern wheels) and L408 E-H (column pattern wheels). The hub for the pattern wheels is shown in 337 L 411.

Since automatic halting of the engine and automatic rewind of the travelling platform are triggered by the end-of-page condition (line pattern-wheel arm at nine o'clock, column pattern-wheel arm at three o'clock), end-of-page was taken as the fixed reference 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 (thirty, fifty, or sixty). 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 (337 L 409, L 411). 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 stereotyping punch wheels (A/173). These constant right-hand margins are 0.75" for large type, and 1.6875" for the small. For single column working these margin constraints still apply i.e. the single column will appear biased 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 X 23, bottom left. The bottom three views show examples of other layouts. The middle view, for example, shows a two-column layout in small type with a narrow column margin.

The various combinations of line and column formats is summarised in the Summary Table in 337 X 23 (bottom right) with a few examples of the layouts in the diagrammatic views. The top half of 337 X 23 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".

Changing Formats

For a complete format change i.e. changing from line-to-line to column-to-column operation as well as line number and number of columns, requires setting a new arrangement of the oscillating bars, selecting the appropriate line and column pattern wheels, manually rewinding to the start position either the line pattern wheel (for column-to-column working) or column pattern wheel (for line-to-line working) and if necessary, adjusting the range of the rewind clutch.

Stepwise procedures for resetting and changing formats are described in the **User Manual (2013)**, **Changing Line Format**, pp. 52-61.

6.4 Additional Modifications

Modifications described above include:

1. Splitting the carriage between large and small matrix pans
2. Addition of two runaway pawls to prevent carriage runaway
3. Modification to the rewind gear to a allow partial variable carriage rewind
4. Redesign of the rewind dog-clutch to prevent jamming.

Several other modifications, precautionary and remedial, were made:

1. Addition of column-to-column dashpot to cushion carriage traverse
2. Additional anchoring of stereotyping table
3. Extensions to increase drop-height for falling weights
4. Provision for clamping the matrix pans
5. Additional guide for stop cord pulley
6. Reset lever for stop weight
7. Addition of uncoupling clutch and auxiliary drive for output apparatus
8. Modifications to framing support.

1. Column-to-column Dashpot

Babbage gives no details of formatting options or of the pattern wheel tooth layout to achieve these. Common preferred formats were identified (337 X 23) and pattern wheels specified to realise them. Balancing various layout combinations with pan sizes and font heights, the maximum line-to-line carriage movement emerges as 0.275”. This is the maximum distance the carriage will advance in one complete calculating cycle to generate one new result.

The falling weight that drives the platform line-to-line ($\mathscr{B}$, A/166 End View) needs to be heavy enough to overcome initial static friction without initial hesitation. However, if heavy enough to do this the carriage will accelerate rapidly until stopped by the next pattern wheel tooth in an uncushioned impact with the line catch. With only 0.275" linear travel between lines, the jarring is tolerable. However, in the case of column-to-column traverse, the carriage travels as much as 5¼" between stops. The uncushioned impact of a heavy carriage (made heavier by splitting into two equal carriages) accelerating over approximately 5"was thought to be excessive and a removable oil-filled dashpot was added to buffer the impact.

The dashpot is fitted under the carriage along the centre line of the carriage frame (Figs. 6.5, 6.24, A/147 Plan) in direct line with the boss of the large idler gear (^{6}***D*** A/163, for assembly detail see 337 L 22, for dashpot parts see 337 L 38). The dashpot acts as a linear speed governor. Its action is unidirectional. In the damped direction (right to left of the carriage A/163) the piston forces oil through an annular aperture. During the return stroke (left to right of the carriage) the piston is driven against a spring and opens the oil-feed aperture to allow undamped movement (337 L 22).

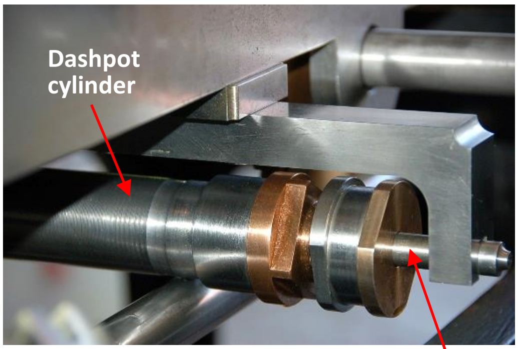


Fig. 6.24: Dashpot.

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.

Stepwise procedures for adjusting the degree of damping are described in the **User Manual (2013), Adjusting Dashpot Damping**, pp. 61-3.

2. Additional Anchoring of the Stereotype Table

The table stand for the stereotyping apparatus falls outside the base rails that run the length of the whole of the Engine (A/163, A/164) so the output apparatus and the calculating section are not resting on the common base provided by the rails and no fixing is shown between the table assembly and the calculating section. Additional fixing was provided by extending the boss of the idler gear (^{6}***D*** A/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 left-to-right tilt. Jacking points on the table legs allow for adjustment and for correction of any tilt left-to-right or front-to-back.

3. Drop-height for Drive Weights

Working back from the carriage to the pattern wheel pulleys for the travelling platform falling weights shows that a pulley rotation of 340° is required for end-to-end travel line-to-line. The corresponding length of the drop for the falling weights $\mathscr{A}$ and $\mathscr{B}$ (A/166 End View) for the 9"-diameter pulley, indicated in A/166, is 26.7" (340/360 x 9π). However, the drop from the underside of the weights to the floor is only about 11". The extra drop is provided by additional pulleys suitably raised (L26). The pulleys are mounted on extension arms fixed to the existing framework (Fig. 4.1).

No detail is given of the form of the falling weights $\mathscr{A}$ and $\mathscr{B}$ (A/166 End View). Driving forces required will depend on friction, traction, load, and eventually wear. Increasing or reducing weight is anyway required to compensate for the presence or absence of one or another of the matrix pans. So that the appropriate weight can be flexibly adjusted, each of the masses $\mathscr{A}$ and $\mathscr{B}$ consist of ten separate weights some of which are C-shaped i.e. a section of the ring missing to provide a radial slot so that they can be stacked and removed with a no dismantling (Fig. 6.25).

Fig. 6.25: Falling weights.

4. Pan clamping

No means of fixing the pans to the carriage is shown whether by screw fitting or wedges (A/147 End View and Elevation). The left-to-right fixing (A/147 Elevation) was retained as a slide fit of the pan into the machined tray in the carriage, and screw clamps were used to locate and fix the trays front-to-back (337 L 25). 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 337 L 363, and cross-sliding guide 337 L 354). The rods and sliding guides are removable loose pieces and the clamps are tightened with knurled nuts (Fig. 6.3).

5. Stop cord pulley

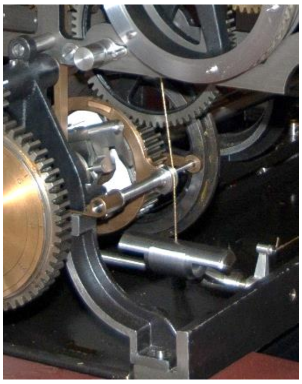


Fig. 6.26: Stop cord routing.

The cord for the stop weight $\mathscr{C}$ (A/166 End View, bottom centre) is shown 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 337 L 27).

6. Stop weight reset lever

The pattern wheels are exposed on the outside of the control mechanism for the stereotyping table (Fig. 4.1, A/163, A/166) and these can conveniently be gripped for manual rewind or by using the additional wooden removable rewind handle. However, the stop pulley (${}^{3}\mathscr{L}_{1}$, A/166 End View red outline) is well buried in the mechanism and finger access for rewinding is restricted. A lever was added to assist with the rewind.

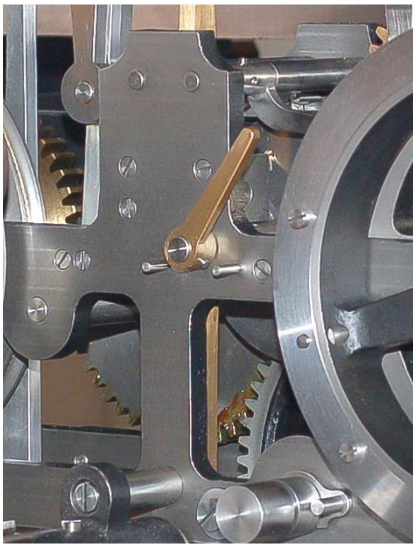


Fig. 6.27: Stop-pulley reset lever.

It is desirable to restrict the rotation of the stop pulley both clockwise and anticlockwise. Anticlockwise restriction prevents the weight being inadvertently lifted above the sides of the trough or gutter during manual rewind, as well as ensuring that the catches and trips remain associated with the appropriate levers. Clockwise restriction prevents the cord going slack and jumping the pulley sidewalls when the stop weight reaches the end of its travel when dropping. A lever with unobstructed access was added 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 nine and three o'clock (Fig. 6.27, 337 L 24).

7. Release clutch and auxiliary drive for output apparatus

The original designs show the output apparatus as a single monolithic 'hardwired' assembly coupled directly to the calculating section. The output apparatus and the calculating section form a single mechanically integral device with both parts driven by a single unbroken shaft, $^{4}\mathscr{C}$, that runs the full length of the underside of the Engine and driven by large bevel gears ($\boldsymbol{A}_0$, $^{4}\mathscr{C}$) under the cam stack (A/163 right bottom). In the original design there is no provision for uncoupling the calculating section from the output apparatus: to run the output apparatus one must run the Engine.

There are build, operational and maintenance reasons for wanting to run the output apparatus and calculating section independently. During building, commissioning, fault-finding and maintenance there are adjustments that require full or partial cycling of the apparatus. Debugging faults often requires inspection of slowed-down motions and the need for small incremental advances. With no facility for driving the printer locally, cycling of the output apparatus requires two people – one at the main Engine crank end and one at the output apparatus opposite, over eleven feet away and unsighted. Advancing the printer by small increments is difficult with this arrangement. With an additional drive local to the output apparatus one person can operate the output apparatus and better control incremental advancement with the mechanisms fully sighted.

There are also operational considerations. During tabulation there are occasions when the output apparatus needs to be cycled without disturbing the rest of the Engine: priming the inking rollers to ensure uniform ink distribution, advancing the stereotyping moving platform so that the trays can be removed for refilling or replacement, are two. In both situations the internal state of the calculating section needs to remain undisturbed. Conversely, there are occasions when the calculating section needs to be run without the output apparatus. Resetting the starting values in the middle of a run, for example.

Provision was made for uncoupling the output apparatus from the calculating section to allow each to be run independently of the other – this by splitting the drive shaft into two, fitting a hand-operated in-line clutch at the join that allows the drive to the output apparatus to be uncoupled from that of the calculating section, and to fit an additional crank handle and a gear drive so that the output apparatus can be driven independently of the main engine crank. The clutch is engaged only when the Engine and output apparatus are run together i.e. for normal operation.

Release clutch

The release coupling takes the form of a dog clutch. The clutch is fitted under the calculating section located towards the output apparatus. An interlock arrangement ensures that the drive can be disengaged and re-engaged if and only if the output apparatus is correctly phased with the calculating section.

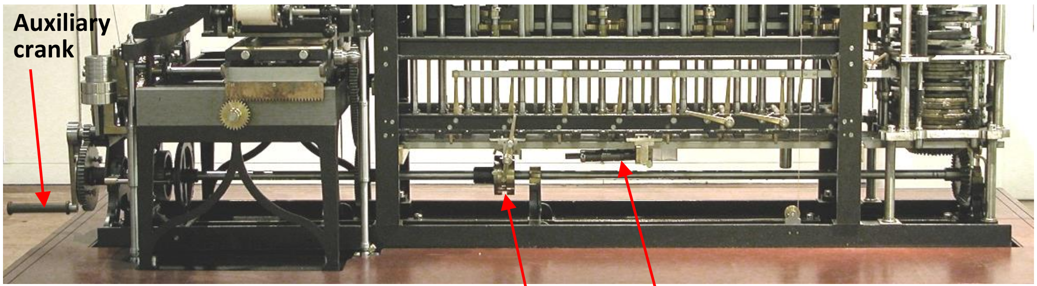


Fig. 6.28: Output apparatus release coupling. Lock counterbalancing springs

The dog clutch consists of a steel disc on the printer side pinned to the main drive shaft, and a disc (bronze) on the main cam-stack side keyed so that it can slide on the shaft. The sliding disc is fitted with two dogs that engage with slots in the fixed disc (337 M 361, M 362). There are three slots in the fixed disc, two opposite each other to receive the drive dogs, and one to receive the rocker lug for locking the output apparatus. There is one slot in the sliding disc to receive the rocker lug to lock the calculating section. The clutch is operated by two levers, one at the front of the Engine (the clutch lever, Fig. 6.31) that engages or disengages the clutch, the other at the rear of the Engine (the locking lever, Figs. 6.29, 6.30) operates a rocker at the top of the clutch that acts as an interlock. The clutch lever (front) engages and disengages the drive by sliding a yolk which slides the dogs into or out of engagement. The clutch is engaged when the clutch lever is inclined to the right

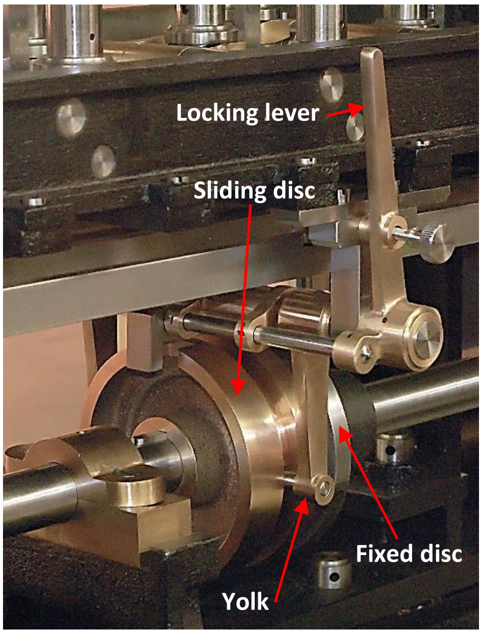


Fig. 6.29: Release coupling. Rear view.

as seen from the front of the Engine (Fig. 31). The lever is held in the engaged or disengaged positions by a knurled thumbscrew which fixes into one of two threaded holes – this to prevent creep or accidental movement. The operation is fail-safe i.e. there is no danger if the thumbscrew is not used to secure the lever.

The locking lever (rear) operates a rocker located at the top of the two discs. The function of the rocker is to ensure that (1) the clutch cannot engage unless the

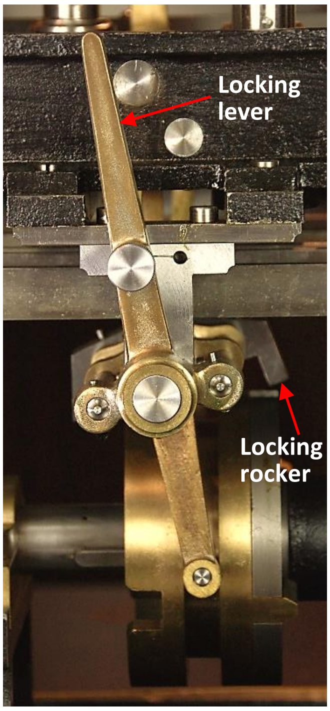


Fig. 6.30: Locking lever (view from rear).

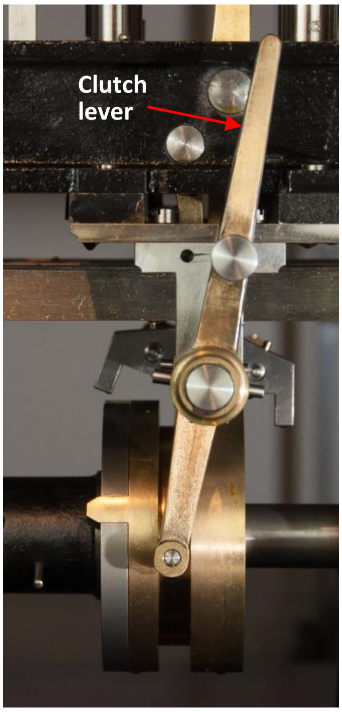


Fig. 6.31: Clutch lever (view from front).

calculating section and output apparatus are both at one, and only one, specific point in the cycle (2) in the uncoupled state the printer is locked when the Engine is free to run, and the Engine is locked when the printer is free to run: in one position the rocker locks the calculating section and frees the output apparatus to turn; in the other position it locks the output apparatus and frees the Engine to turn. The locking lever can be fixed in one or another of the positions with a knurled screw as for the engagement lever. For normal operation i.e. with the output apparatus coupled, both levers face to the right as seen from the front of the Engine.

The Engine is locked at 35 units (252°) for the output apparatus to be run independently (Timing Diagrams F/385/1 and 337 X 21). The reason for choosing 35 units for the uncoupling point is that in the event that the output apparatus requires attention during tabulation, the run of calculations can be intermitted with the current result secured, the output apparatus can be attended to by running it using the auxiliary crank, and the tabulation resumed with no interruption to the sequence of results. More specifically, at 35 units the result of the current calculation is set and locked in the results column, the result has been transferred to the print wheels and stereotyping punch wheels but no action has yet been taken to print. When the printer is run using the auxiliary crank (i.e. independently of the calculating section) the positions of the printing wheels and stereotyping punch wheels remain set with the result currently locked on the results column. After the output apparatus has been run using the auxiliary crank it is advanced (using the auxiliary crank) back to 35 units for recoupling after which the printing cycle can resume with no disruption to the tabulation sequence.

When the engine is run independently the output apparatus is locked at 0 units (337 X 21). The reason for this relates to the functioning of vertical lock ($^{5}\mathscr{L}$, A/174 Locking Bar; also A/176, Fig. 4.7) that operates on the compound racks ($_{n}{}^{6}\boldsymbol{R}$ A/174) near the results column. The compound racks are directly coupled to the last sectors and, in general, move each calculating cycle whenever the Engine is run. The vertical lock for the compound racks is driven by a cam in the output apparatus. If the Engine is to be run independently of the output apparatus then the compound racks must be in their unlocked state to prevent the mechanism jamming i.e. the output apparatus must be locked at a point in the cycle where the compound rack locks are disengaged and the compound racks free to move. Zero units was chosen as an appropriate and convenient point to lock the output apparatus. This is not the only time the racks are unlocked, but 0 is simple and convenient.

Detailed procedures are given in the **User Manual (2013)**, **Uncoupling the Drive,** pp. 64-9 that describe the clutch lever settings to run the output apparatus and calculating section separately, and for recoupling for normal use where the calculating section and output apparatus operate together.

The following table summarises the lever positions, cycle points, and functional status of the Engine and output apparatus when run separately and together.

In the table ‘right’ and ‘left’ are as seen facing the clutch lever and locking lever as though operating them. So moving the locking lever (rear of Engine) anti-clockwise moves it from facing right to facing left as seen while facing the lever.

STATUS TABLE				
Function	**Cycle Units**	**Clutch Lever (Front)**	**Locking Lever (Rear)**	**Unit Locked**
Run Printer Only	35	Left	Left	Engine
Run Engine Only	0	Left	Right	Printer
Run Both	35	Right	Left	Neither
Run Neither (Both Locked)	0	Right	Right	Both

Debugging using the Release Clutch

Use of the release clutch as a debugging aid for faults or jams that occur is limited. The interlock mechanism of the clutch prevents the calculating section and output apparatus being run independently at the same time. More importantly, uncoupling can only take place at two specific points in the cycle, 0 and 35 units. Since it is only by happenstance that a jam, for example, would occur at either 0 or 35 units It follows that the clutch cannot in general be used to uncouple the two sections of the machine to isolate them for the purposes of debugging a jam as it would not be possible to cycle the Engine to the uncoupling points.

Auxiliary Crank

An additional crank was provided to drive the output apparatus when uncoupled from the calculating section (Fig. 6.1, p. 100; Fig. 4.1, p. 49). The crank, pinion and 4:1 reduction gear are duplicates of the main drive handle at the main cam-stack end. The drive pinion of the main crank is below the drive gear. In the auxiliary crank it is above (see 337 M 34 for detail). A chapter disc and pointer to indicate the current point in the cycle were fitted in the same style as that on the main drive crank. The auxiliary chapter disc is engraved to indicate key events in the printing cycle particularly points in the cycle at which the output apparatus and calculating section can be coupled and uncoupled to ensure correctly phased engagement. So the engraved annotations differ on the two chapter discs. The position of the crank arm is lower than that of the main crank handle and therefore less convenient to operate. However, it does not obstruct access to the pattern wheels or stereotyping trays.

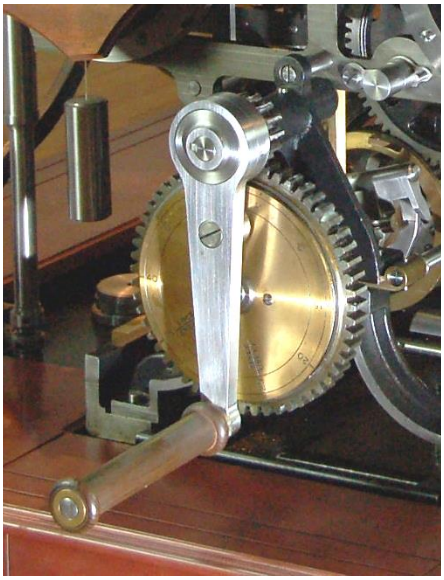


Fig. 6.32: Auxiliary crank.

The main shaft bearing support as originally designed is shown in drawing A/165 top left. An additional cast bearing (337 M 342) for the crank handle shaft and drive pinion was piggy-backed onto the original bronze bearing (337 M 351 & 2 for drive pinion detail and pinion shaft). The main shaft was extended to accommodate the drive gear. The insertion of the clutch in the main drive shaft did not provide sufficient extension of the shaft for the handle and drive gear and a new section of shaft was made.

The crank has a ratchet that allows only unidirectional drive (337 M 345 for ratchet detail; M 346 for pawl detail). The ratchet follows the same design as that of the main drive with the difference that the ratchet on the auxiliary crank can be disengaged to free the handle and also to prevent free-wheel clicking when the Engine is driven from the main crank.

The auxiliary crank is disengaged by loosening two knurled nuts at the back of the large driven gear and rotating ratchet cover plate 20° anticlockwise (as seen by the operator facing the chapter disc) (337 M 357 for detail). The clockwise position of the cover moves a pin which keeps the sprung pawl engaged. Rotating anti-clockwise releases the pin, and the pawl is free to drop out putting the drive in neutral. The pin does not actively disengage the pawl: it frees the pawl to drop out of engagement if it stops at or near bottom dead centre. If the ratchet pawl does not drop out to put the drive in neutral, backing off the crank (anticlockwise) a few degrees will pop out the pawl and disengage the drive.

It is possible to drive the Engine from the output apparatus crank but it is not advisable to try: the output apparatus drive shaft has a smaller cross-section that the main cam-stack shaft and the principle of smaller shafts driving larger ones is not sound. There is also the risk of 'wind up' i.e. torsional loads producing twists in the drive shaft which would affect timing. There is no ratchet release mechanism for the main crank.

The order in which the modification was implemented was to leave the original drive shaft intact and assemble the main shaft end bearing as originally designed. This is to ensure exact alignment of the end bearing. Only when the end bearing was fitted and aligned was the drive shaft severed to fit the clutch and an extended shaft added to accommodate the hand crank and drive.

8. Modifications to Frame

The printer apparatus (the upper box-assembly of the output apparatus (Fig. 2.3, p. 12), A/163) is fixed to the vertical frame by four tapped fixings with the top two fixings in tension taking most of the load. The original designs show no fixing provision other than bolting to the frame. During assembly there was concern that these fixings would be insufficient for the weight of the printing apparatus. The weight of the stereotyping table below was less of a concern: it is a free-standing assembly supported by its own legs and tethered to the frame with fixings that are adequate.

Provision was made to provide additional support for the printing apparatus to relieve the fixings to the upper section. Four additional legs were provided (Fig. 6.33). One pair supports the rear of the box frame i.e. where it abuts the main vertical frame of the calculating section (Fig. 6.1, p. 100; Fig. 6.33, p. 147). Two heavy curved brackets were added to the front framing and are supported by a fish-belly cross-member. The fish-belly cross-member is supported by two additional legs, one at each end. The four legs are turned steel with bright machine finish and period beading. The brackets and fish-belly cross-piece are black cast iron. Bright finish on the turned steel legs was chosen in preference to cast iron so as not to compete with the curved bracing of the table legs ([1]**A**, A/163) in the original design, and a falling rather than a rising curve chosen for the two brackets in preference to a rising curve as more in keeping with contemporary practice (Fig. 6.33). The four additional legs are fitted with jacking screws for levelling and small mounting plates.

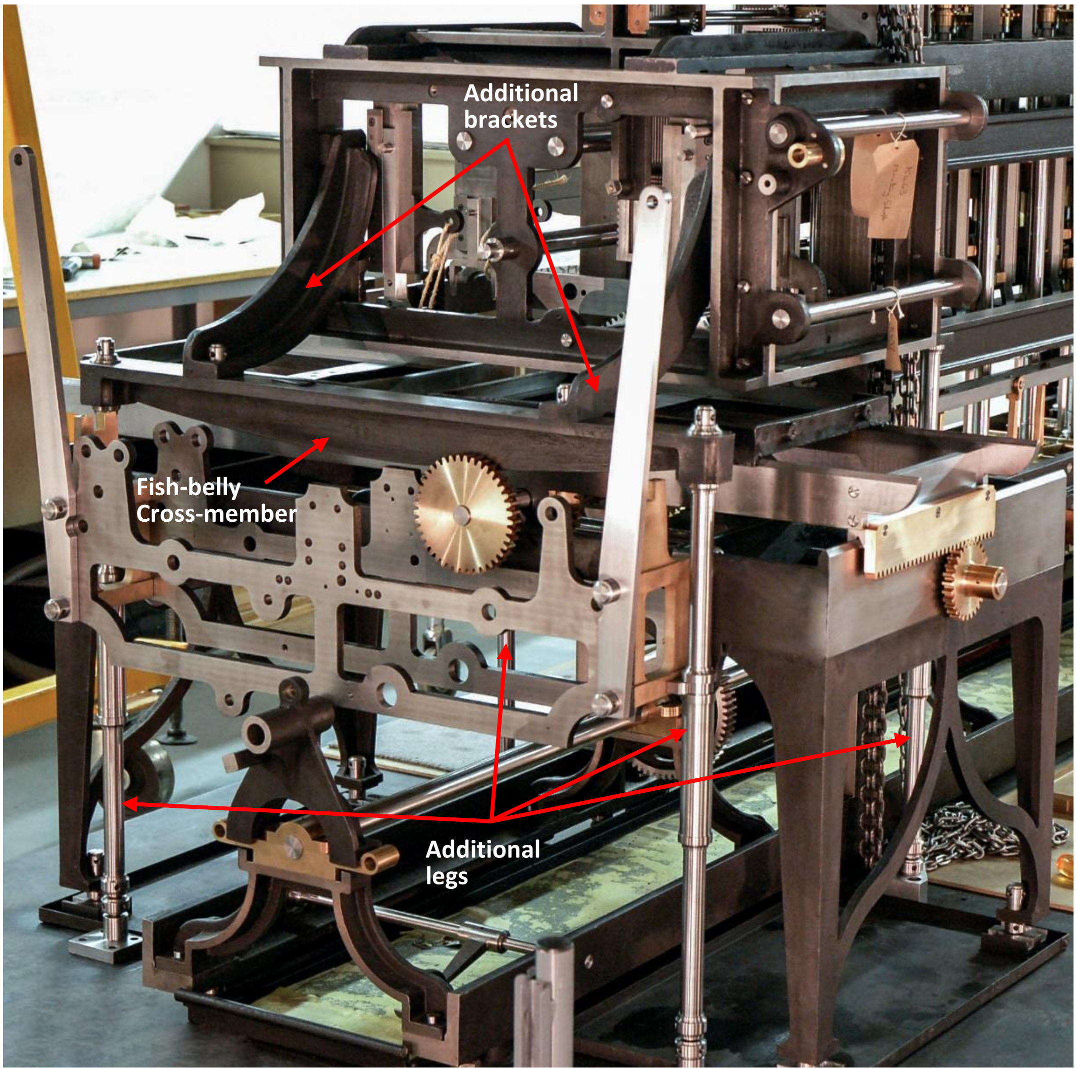


Fig. 6.33: Modifications to output apparatus support framing (during construction).

7. Drive

This section describes the mechanisms for transferring power from the hand crank, the first mover of the whole Engine, to the main cams and the mechanisms for producing the lifting and turning motions of the axes to perform the repeated additions required for tabulation by the method of differences. It also describes the main overall drive train and the automatic halting feature.

The prime mover is a human operator turning the main crank handle ($^{8}\mathscr{B}$, A/163 main Elevation; Figs. 2.1, p.11; Fig. 3.1, p. 19). Power is transferred from the hand crank to the cam stack drive shaft (***B***) via bevel gears ($^{7}\mathscr{M}$, ***C*** A/163 right).

The cam stack consists of a vertical tier of twenty-eight conjugate cams, keyed to be main vertical shaft, which orchestrate the vertical and circular motions of the figure wheel, sector, warning and carry axes in the adjacent calculating section. Circular motions are derived from twelve cams in the upper section of the cam stack i.e. six pairs of conjugate cams $^{18}\boldsymbol{A}$ through $^{28}\boldsymbol{A}$, A/163. The vertical motion cams are the sixteen cams in the lower section of the cam stack $^{1}\boldsymbol{A}$ through $^{16}\boldsymbol{A}$ with $^{1}\boldsymbol{A}$ at the bottom (A/163). The cam between $^{20}\boldsymbol{A}$ through $^{21}\boldsymbol{A}$ is not labelled and its number omitted in the sequence. This is probably to correct for the fact that if the cam numbering is to follow a strictly monotone increasing sequence then the lowest cam in the top upper set of twelve cams should start with $^{17}\boldsymbol{A}$ not $^{18}\boldsymbol{A}$ as annotated. This is corrected in A/160 right (see table **Circular Motion Cams**, p. 160 for renumbering). In the upper set of twelve cams $^{18}\boldsymbol{A}$ and $^{19}\boldsymbol{A}$ form a pair, for example, as do $^{27}\boldsymbol{A}$ and $^{28}\boldsymbol{A}$ i.e. the cams shown closer together belong to different pairs: the conjugate pairs are those further apart.

The mechanisms that translate the rotation of the cams into vertical and circular motions are those shown in skeleton in the bloc below the eight figure wheel columns i.e. to the left alongside the cam stack in A/163.

The circular motions for the figure wheel, sector and warning axes are intermittent and reciprocating (bidirectional) consisting of a sweep through a fixed angle (a sector of a circle) for the drive stroke and, presently, a return stroke. These motions are intermittent and phased to occur at specific parts of each calculating cycle (337 X 21). The cams and cam-followers translate the rotation of the cams into linear reciprocating motion of drive links which drive sliding racks. The linear reciprocating motion of the racks is translated into circular reciprocating motion driving the figure wheel, sector and warning axes by sector pinions fixed to the base of the axes as shown in A/171 bottom right.

The circular motion for the carry axes is derived from a separate mechanism and not from cams in the main cam stack. The mechanism, of which gears $^{7}\mathscr{P}$ and $^{7}\mathscr{N}$ are part, is shown in A/163 alongside the right hand vertical framing member ($^{3}\mathbf{N}_{2}$). The mechanism is driven by a sleeve shaft $^{7}\mathscr{V}$ and the output is intermittent circular motion of horizontal shaft $^{2}\boldsymbol{B}$ from which the carry axes $^{1}\boldsymbol{C}^{0}$ through $^{1}\boldsymbol{C}^{6}$ are driven via bevel gears $^{2}\boldsymbol{S}^{0}$ through $^{2}\boldsymbol{S}^{6}$ (A/163).

Vertical motions for the figure wheel, sector wheel and warning axes are generated from the linear reciprocating motions of horizontal bars, themselves driven by the cams and cam followers in the main cam stack. The general principle is that linear reciprocating motion of the bars, which have slots, drive bell cranks to lift and lower the axes. An indicative example is the figure wheel axis ($^{2}\boldsymbol{T}$) bell crank and slotted bar ($^{12}\boldsymbol{A}$) in A/163 (Fig. 7.1).

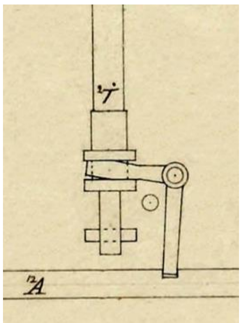


Fig. 7.1: Bell crank, even differences (A/163) (detail).

Interposed between the crank, $^{8}\mathscr{B}$, and the drive shaft bearing ($^{5}\mathscr{F}$, $^{6}\mathscr{F}$) is a ratchet drive ($^{8}\mathscr{C}$) that ensures that the main drive cannot be reversed i.e. that positive drive is unidirectional with the handle free to rotate in reverse but without driving the shaft. The ratchet is coupled to the bevel gear via a rocker clutch ($^{3}\mathscr{K}$, $^{9}\mathscr{D}$) that is disengaged by a gut cable (ℓ) running from the output apparatus via a series of pulleys. The clutch is activated by the stereotyping apparatus at the end of a page or when the stereotyping trays have received a full complement of results. Activating the clutch uncouples the drive and the Engine halts allowing the matrix pans to be renewed to receive new results when the calculation resumes.

A status plate or chapter disc and pointer were added to indicate where in the cycle the Engine is at any time. The brass disc is fixed to the phasing gear ($^{7}\mathscr{N}$ A/163), which is in direct sight of the operator, and is engraved with divisions every five units (0, 5, 15 etc.) using Babbage's fifty-unit cycle convention where each five-unit interval corresponds to 36°. A single rotation of the phasing gear corresponds to one full calculating cycle. Key events are engraved with annotations 'Full Cycle' (zero), 'Set Odd' (20 units), 'Half Cycle' (25 units) and 'Set Even' (45 units). A swan-necked pointer is fixed to the left front upright, as seen facing the crank, and acts as a datum. Status indication is indispensable during the procedures for setting initial conditions where, for example, the Engine needs to be advanced to 10 and 35, and then

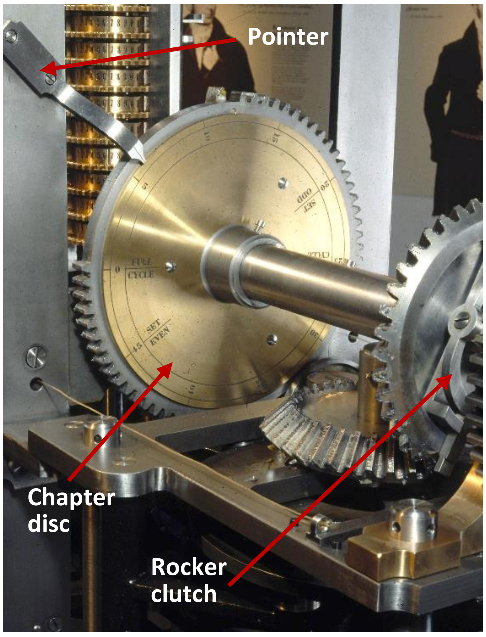


Fig. 7.2: Chapter wheel and pointer.

20 and 45 units when setting initial differences on the figure wheel axes at the start of a calculation. Status indication is also indispensable for debugging – indicating the point in the cycle of a jam, for example. There is no provision in evidence in the original design for indicating where in the cycle the Engine is at any given time.

On the underside of the cam stack is a second pair of large bevel gears ($\boldsymbol{A}_0$, $^{4}\mathscr{C}$, A/167) driven by the main cam stack shaft, $\boldsymbol{B}$. These drive the output apparatus via drive shaft, $^{4}\mathscr{C}$, which runs the length of the underside of the engine (A/163). There is a separate set of cams local to the output apparatus that controls the internal operations and timing of the printing and stereotyping apparatuses (see **Cams, Drive and Control**, p. 96).

Main Drawings: A/159, A/163.
Related Drawings: A/160, A/164, A/165, A/167.

7.1 Main Crank and Ratchet Drive

The crank consists of a drive arm ($^{8}\mathscr{B}$, A/163) and a drive handle. The handle is made from lignum vitae which, though not specified, was regarded as typical of the time. 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. The length and thickness of the arm are taken from A/163 and A/164. There is no end elevation of the crank so the intended shape of the arm is not known. The final form was taken as in keeping with contemporary practice.

The ratchet prevents the engine being driven in reverse which would damage the calculating mechanism, and allows the operator to best position the handle for controlled effort to cycle the Engine from standstill. The ratchet also allows the operator to nudge the Engine, by increments, to a particular point in the calculation cycle by repeated short controlled pulling movements (see **User Manual (2013)**, **Operating the Engine**, p. 15-19). This is especially useful during the procedure for setting up initial values, adjustment and fault-finding. The position that most favours controlled turning or incremental nudging is with the handle at waist-height at the start i.e. with the handle at the lowest point, and drawn towards the operator by pulling. (Operator's left hand is closest to the cam stack.)

The ratchet mechanism consists (A/159, end view and section) of a ratchet wheel, $^{1}\mathscr{D}$, and matching pawl, $^{2}\mathscr{E}$, housed in a pawl wheel which acts as its casing, $^{8}\mathscr{C}$. A leaf spring fixed to the pawl acts against the inside of the case to bias the pawl. The pivoted end of the pawl is tucked against a machined pocket in the side wall of the case and this takes the load rather than the pawl pivot which is a loose fit. The pawl is sandwiched in place by the pawl wheel cover ($^{8}\mathscr{I}$) which is rebated into the case and secured with counterbored cover screws (A/159 shows the case with the cover removed but the fixing screws fitted.).

Counterbores for the pawl wheel cover fixing screws have insufficient clearance from the wall of the casing and the section view (A/159) shows the cover fixing screws breaking through the back of the case. Fixing screws were 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.

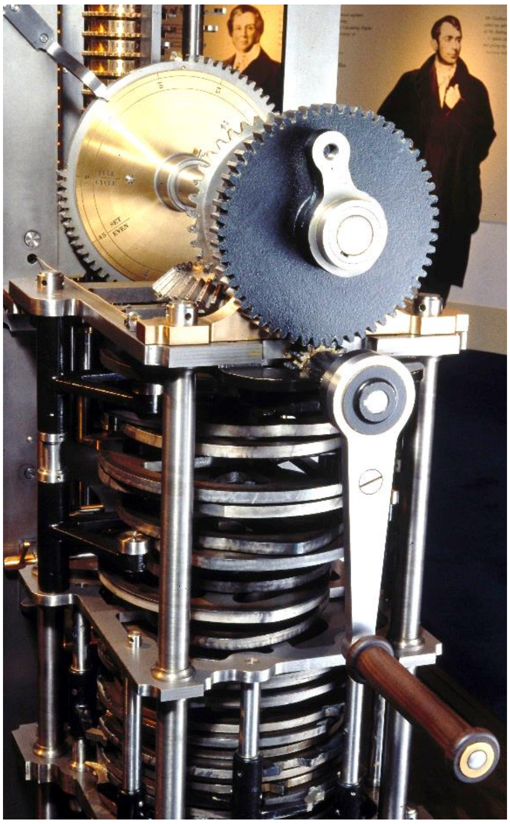


Fig. 7.3: Main crank and reduction drive.

The outer circumference of the case was extended to take 56 gear teeth to allow a 4:1 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 14-teeth pinion (337 D 323), a crank (337 D 322) and a horseshoe mounting (337 D 321) for the pinion (Fig. 7.9). The crank, to which the crank arm is fixed, is a repeat of the wheel crank above the handle shown in Fig. 7.3. The boss of the crank is keyed to the pinion, and the drive arm is fixed to the crank with a shoulder screw (337 D 327). 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 originally intended and as shown in A/163. With the reduction gearing the Engine is four times easier to turn but runs four times slower. With the 4:1 gearing the average time for one calculating cycle is 6 seconds i.e. ten results per minute.

Without the reduction gears the engine is driven by turning the handle anticlockwise as seen facing the crank. The addition of the reduction gearing reverses the direction of the drive. The main loads on the drive occur as brief shock loads at the start of the cycle (unit zero) and at the half-cycle point (unit 25). These spike loads are caused by all the figure wheel locks operating simultaneously (Timing Diagram 337 X 21) creating two short peak loads in each full cycle. Less severe point-loading occurs 10 and 35 units when two sets of locks operate, but not simultaneously.

As mentioned, for the convenience of the operator the handle is best positioned at the lowest point (roughly waist height) for a controlled pulling action to overcome the start-of-cycle load. Without the reduction gears the half-cycle load would occur at the top of the arc requiring a pushing action which is awkward and might reasonably have encouraged Babbage to fit a reduction gear of his own. So an additional advantage of the reduction gear 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 controllable than a pull at the low point of the crank, and a push at the top as would be the case with a single-turn cycle. The calculating mechanism is susceptible to jamming if the drive is not steady, firm and at constant speed. The mechanism is especially susceptible to jamming if the transitions through the points of shock loads is hesitant. Anything that assists in achieving even drive reduces the risk of jams so being able to pull the handle at its lowest point to ride the vulnerable time windows is a more than trivial benefit.

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 in this way because of the load and the awkwardness of the movement. For many jams simply releasing tension by backing off (i.e. with minimal rotation) is sufficient. No provision in the original design was made to disable the ratchet drive to provide positive drive in the reverse direction and none was introduced, as deliberate or inadvertent reverse drive at an inappropriate point in the calculating cycle runs the risk of damage to the calculating mechanism.

Lubricating the Cam Stack Shaft Bearings

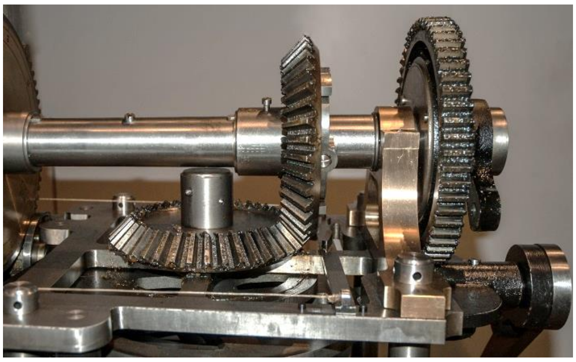


Fig. 7.4: Cam stack bevel-gear drive.

The cam stack drive shaft (***B***, A/163) is held at the top by a journal bearing in the upper framing plate ($^{8}\mathscr{F}$, Fig. 7.4) and is supported below by a thrust bearing in the lower framing plate $^{5}\mathscr{B}$ (A/160 and A/163). Detail of the thrust bearing is shown in A/160 lower right. The bearing consists of an inverted cup-shaped collar, keyed to the shaft, resting in an annular oil bath rebated into the lower framing plate, $^{5}\mathscr{B}$ (Fig. 7.5). No provision in the original drawings 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 (***C***) obstructs access to the upper journal bearing.

Provision was made for lubrication: oil is supplied to the bearings through an oiling hole in the upper bevel gear (***C***) and conveyed along the length of the cam stack 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 ($^{8}\mathscr{F}$) under the bevel gear (***C***). 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 cut-outs (D373) were machined into the thrust face of the collar. For oiling procedure see **User Manual (2013)**, **Lubrication**, p. 96).

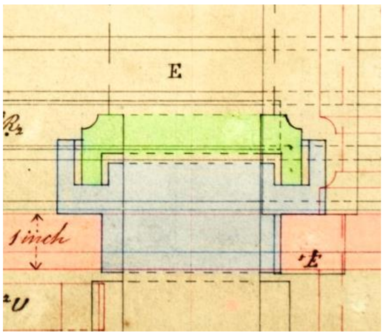


Fig. 7.5: Cam-stack thrust bearing (A/160) (colour added).

7.2 Uncoupling Clutch: Automatic Halting

The Engine is automatically halted when the stereotyping apparatus reaches the end-of-page i.e. when the stereotyping trays have received a full complement of results. The control mechanisms for the detection of end-of-page are described in Chapter 6, p. 127-. Briefly the end-of-page condition triggers the release of a falling weight ($\mathscr{C}$, A/163 bottom left, Fig: 6.20) in the stereotyping apparatus. The weight operates a trip lever (***B***) that pulls a gut cord (***ℓ***) running from the trip lever at printer end of the Engine, through a series of pulleys, to operate a rocker clutch ($^{3}\mathscr{K}$, $^{9}\mathscr{D}$ A/163) at the main crank.

Activating the clutch uncouples the drive i.e. when operated, the clutch breaks the drive between the crank handle and the bevel gear ($^{7}\mathscr{M}$, ***C*** A/163). The Engine halts and the crank handle immediately runs free. With the Engine stationary in the halted state the matrix pans can be maneuvered into positions from which they can be removed and renewed with no interruption to the correct sequence of results that follow.

The clutch is shown in end view in A/159 lower left (Fig. 7.7), and in sectional plan centre left (Fig. 7.6). The clutch rocker ($^3\mathscr{K}$) has a short lug and an extended lug protruding from each of the narrow the ends of the basic oval shape of the rocker. The lugs insert into pairs of slots machined into the rim which stand proud of the clutch body, $^1\mathscr{J}$. The clutch body, $^1\mathscr{J}$, rotates with the oval rocker, itself driven by the main crank shaft $^1\mathscr{A}$. The extended lug (left in Fig. 7.7) protrudes beyond the clutch body $^1\mathscr{J}$ and, with the clutch engaged, inserts between a pair of lugs on the clutch bevel gear, $^7\mathscr{M}$. (working points $^3\mathscr{S}$ -- $^7\mathscr{S}$, A/159 end view) to drive the bevel gear. The rocker has two pivots that allow it

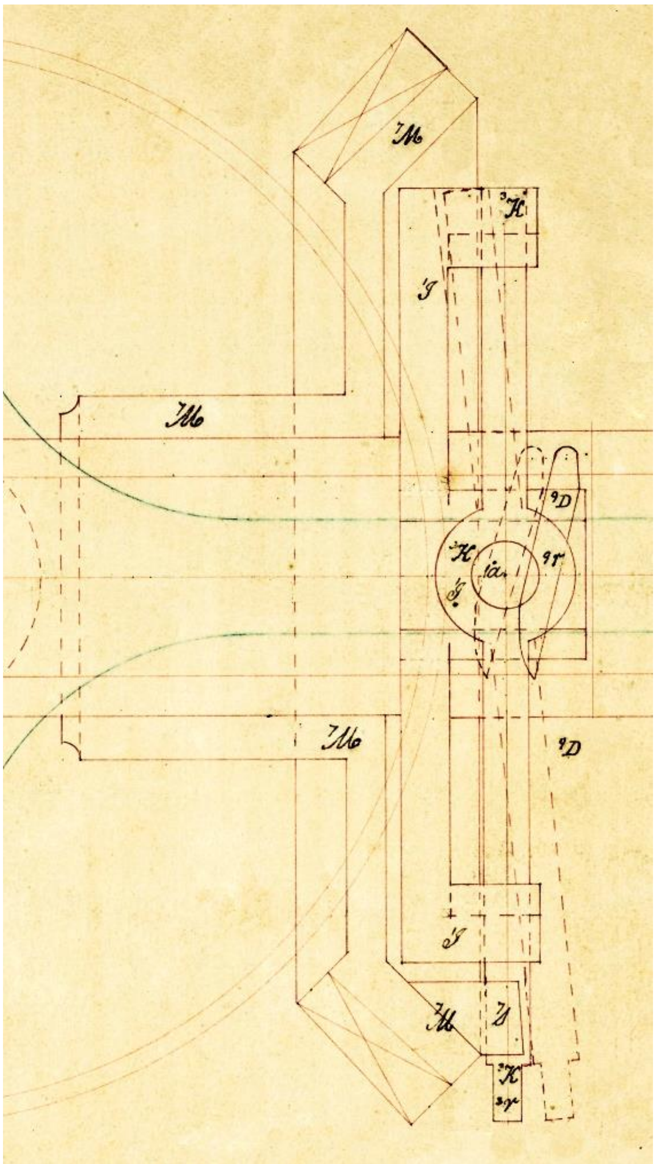


Fig. 7.6: Uncoupling clutch (A/159, Plan).

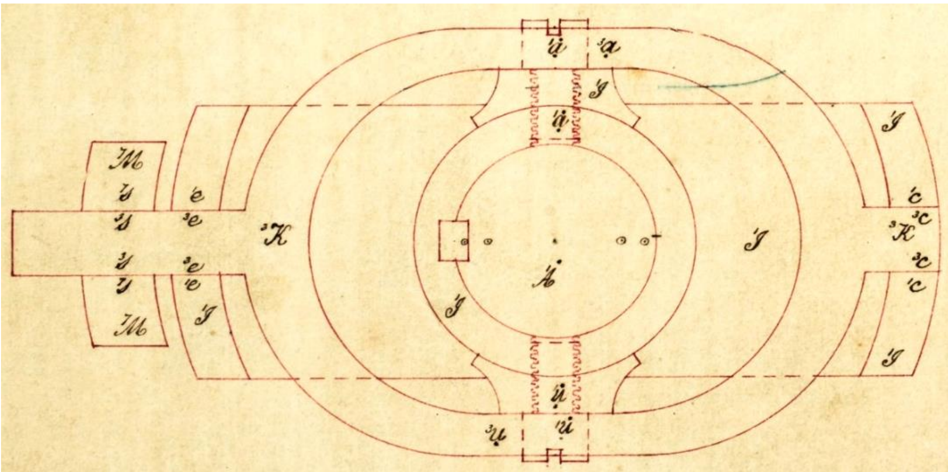


Fig. 7.7: Uncoupling clutch (A/159, End View).

rocker has two pivots that allow it to flip between engagement and disengagement. The clutch is engaged when the extended lug of the clutch rocker is between the bevel gear lugs as shown with the rocker vertical (A/159 sectional plan, unbroken outline, Fig. 7.6), and disengaged when the extended lug is clear of the bevel gear lugs (shown inclined position in dotted outline). With the clutch engaged the drive train consists of the crank ($^{8}\mathscr{B}$, A/163) driving shaft ($^{1}\mathscr{A}$) via ratchet ($^{8}\mathscr{C}$); the rocker ($^{3}\mathscr{K}$) keyed to $^{1}\mathscr{A}$, drives the main bevel gear ($^{7}\mathscr{M}$.) via the extended rocker lug positioned between the two bevel gear lugs. Since the drive is in one direction only the trailing bevel gear lug was dispensed (Fig. 7.8) with and the clutch rocker no longer needs to be aligned with the gap between the lugs to re-engage.

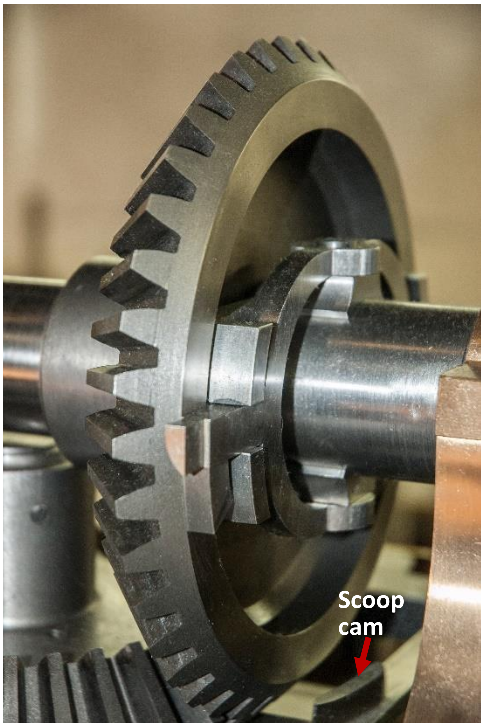


Fig. 7.8: Uncoupling clutch mechanism.

The drive is disengaged by the operation of a scoop lever ($^{9}\boldsymbol{D}$) located on the cam-stack top framing plate (Fig. 7.9). Mounted on the lever is a lozenge (scoop cam) that throws the rocker out of engagement i.e. drives the extended rocker lug clear of the bevel gear drive lug (lever $^{9}\boldsymbol{D}$ is not shown in A/159 sectional plan but is shown in section in A/163).

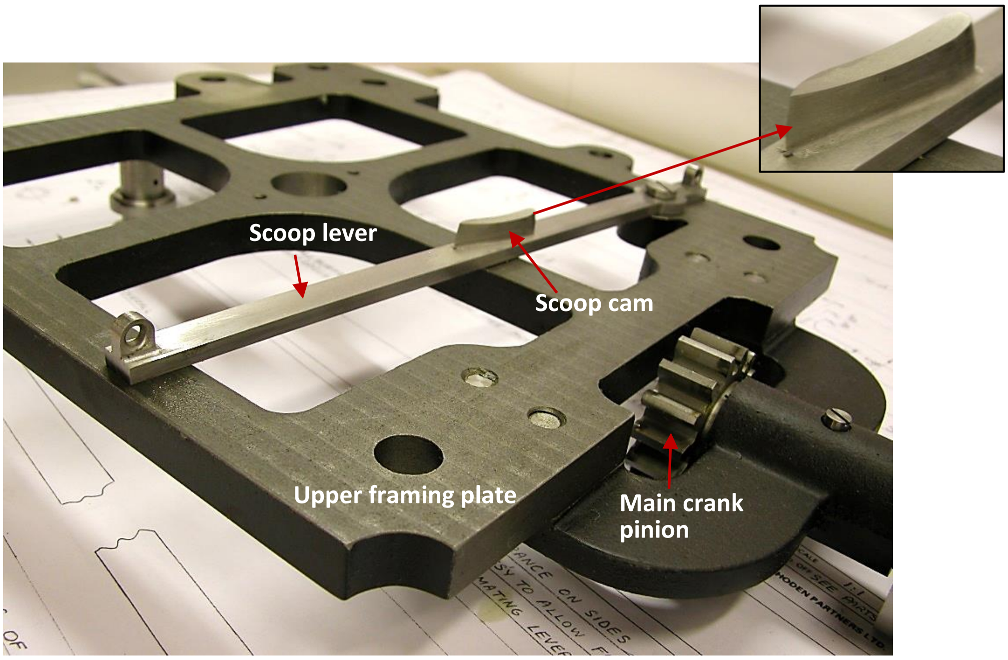


Fig. 7.9: Uncoupling clutch scoop lever, framing plate, and pinion drive.

To disengage the drive, the scoop lever ($^{9}\boldsymbol{D}$), operated by the gut cord from the stereotyping apparatus, moves the scoop cam from the solid to the dotted position (A/159 sectional plan). While the Engine is being driven the bevel gear ($^{7}\mathscr{M}$) rotates anticlockwise as viewed facing the crank (as in Fig. 7.8). With the clutch engaged (rocker position vertical in A/159 sectional plan) the extended lug passes freely on the inside of the scoop cam (bottom-to-top in A/159 sectional plan) i.e. the angled face of the lozenge is outside the path of the extended lug and there is no consequent action. The action of the lever being pulled in by the gut cord positions the scoop cam in the path of the rocker's extended lug and when this lug next traverses the bottom of its trajectory it engages with the inclined face of the scoop cam, drives the scoop lever fully home, and is itself then driven out of engagement i.e. clear of the bevel gear lugs, so breaking the drive train. The Engine halts abruptly and the crank handle runs free.

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 long gut cord jumping its pulleys. The counterbalance also resists the tendency for the scoop lever to be pulled in unintentionally by the trip 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 hanging weight suspended over a pulley fitted to a rear framing member (Fig. 7.10). The pulley is a repeat of that for the clutch cable at the front of the Engine.

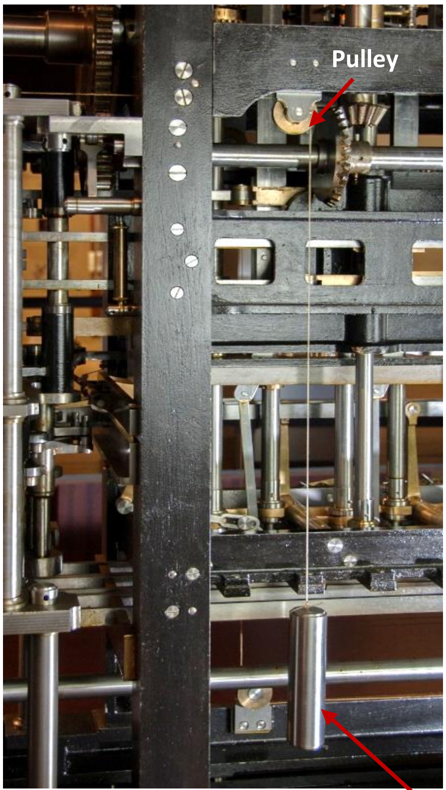


Fig. 7.10: Scoop lever counterweight. (view from rear of Engine).

7.3 Circular Motions

The circular motions required for the figure wheel, sector, warning and carry axes for repeated addition by the method of differences are described in Chapter 3. The circular motions are executed in turn by the odd and even axes in a phased sequence described in the timing notation (F/385/1) for the original design, and in a more detailed version for the engine as built, 337 X 21. This section describes the drive mechanisms that generate the required motions.

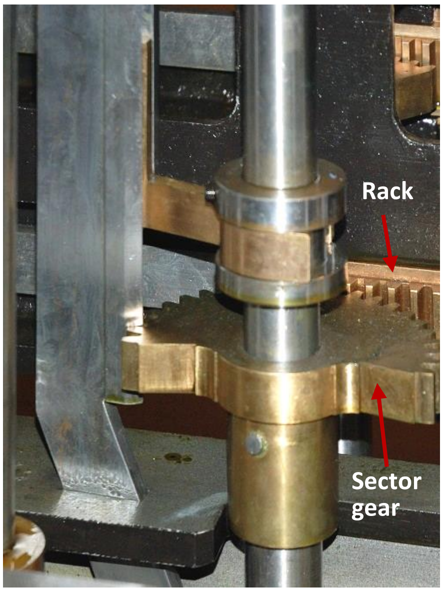
During addition the higher order difference figure-wheels, driven by internal drive arms keyed to the figure wheel axes, are reduced to zero during giving-off. The vertical motion mechanisms then raise the figure wheel axis which disengages the drive arms from the internal lugs by lifting them clear. In the raised position the figure wheel axes return the internal drive arms to the home position until lowered to reengage at the start of the next calculating cycle. The circular motions therefore consist 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 the number given-off to the figure wheels that were its source. During giving-off the sectors are driven by the figure wheels. When restoring, the sectors are driven by restoring arms bearing against internal drive lugs, and the sectors drive the freely rotating figure wheels to restore the number registered before reduction to zero. The restoring arms return to the home position ready for the next cycle. The circular motions of the sector axes therefore 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. These reciprocating circular actions are intermittent and phased to occur at the appropriate points in the timing cycle.

The final phase of the carriage of tens is executed by helices of rotating carry arms on carry axes. The circular motions of the carry axes are not derived from the circular-motion cams in the cam stack but by a separate intermittent drive mechanism ($^{7}\mathscr{P}$, $^{7}\mathscr{N}$, $^{2}\boldsymbol{G}$, $^{2}\boldsymbol{F}$ A/163) coupled to the main drive by shaft $^{1}\mathscr{A}$.

Apart from the intermittent drive for the carry axes, circular motions are derived from six pairs of conjugate cams located on the upper section of the cam stack (A/163). These twelve cams drive cam-followers that rock on fixed pivot shafts (A/170). The arrangement converts the continuous circular motion of the cams to rotational reciprocating motion of the follower arms. The six sets of follower arms drive links to long backing bars to which sections of racks are fixed. Machined channels in the framing pieces form the slides for the rack assembly (Figs. 7.11, 7.12). The racks mesh with sector gears which convert the linear reciprocating motions into the required intermittent reciprocating circular motions for the figure wheel, sector, and warning axes (A/171 lower right).



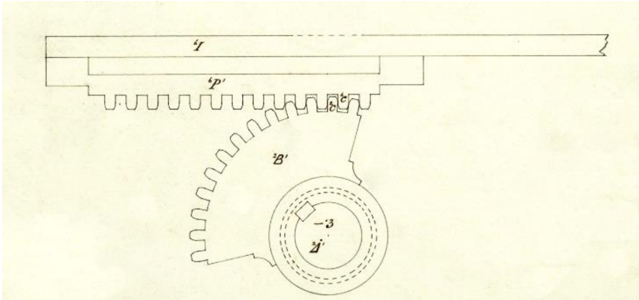


Fig. 7.11: Circular motion rack and sector gear. Figure wheel axis (A/171) (detail).

Fig. 7.12: Circular motion rack and sector gear for figure wheel axis.

The distribution of the racks is best seen in A/163 (see also 337 B 22, and Construction Drawings, Miscellaneous, Unknown 13.jpg). The racks for the four odd difference figure wheel axes are ${}^6\boldsymbol{P}^1$, ${}^6\boldsymbol{P}^3$, ${}^6\boldsymbol{P}^5$, ${}^6\boldsymbol{P}^7$ and the associated rack segments are ${}^2\boldsymbol{B}^1$,${}^2\boldsymbol{B}^3$,${}^2\boldsymbol{B}^5$,${}^2\boldsymbol{B}^7$ (A/163). The four even difference figure wheel rack/sector pairs are in the run immediately above (A/163). The sector wheel and warning axes circular motions have similar arrangements (A/160) though not shown on A/163.

Main Drawings: A/159 left, A/170, A/160, A/161 extreme right, A/169, 337 B 22.

The general layout is given in plan in A/159 left which shows a set of three follower arms on fixed pivots, and partial cam profiles. The pivot ($\mathcal{J}$) for the sector axes follower (${}^7\mathscr{W}$) is shown at ten-to-twelve, the warning axes follower pivot ($\mathbf{K}$) at five-past-twelve, and the figure wheel axes follower pivot (${}^1\mathbf{H}$) at twenty-five-to with follower ${}^6\boldsymbol{J}$. Except for a rotational offset of the cams, which staggers the timing, the layout is identical for odd and even motions and A/159 shows only one set cams and followers i.e. the arrangement is duplicated for the alternate set but is not shown. Followers for the omitted cams use the same pivot shafts. A/170 is a clearer version showing, in one view on the right, the full cam profiles for the even difference sector axes cam pair ($^{27}\boldsymbol{A}$, $^{28}\boldsymbol{A}$) and followers, with the arrangement for the other two motions (figure wheel axes, warning axes) shown separately on the left. The view on the right corresponds to the plan view of the uppermost two cams in the stack (see cam list, p. 160).

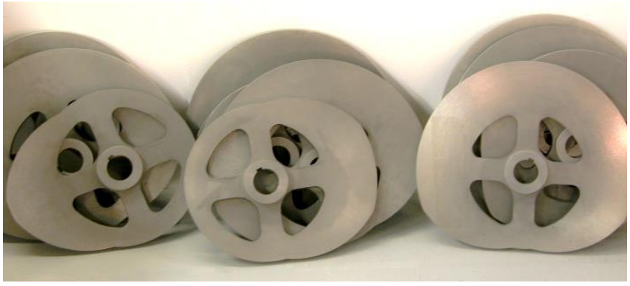


Fig. 7.13: Miscellaneous loose cams.

It is not immediately clear whether the layout shown on A/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 ($^{7}\mathscr{W}^{1}$, $^{7}\mathscr{W}^{2}$ and $^{6}\boldsymbol{J}^{2}$, $^{6}\boldsymbol{J}^{1}$ A/159 left), for example, are in different planes but are both drawn with unbroken lines and superimposed. There are other signs of draughting inconsistencies: the two arms of the same figure wheel followers are shown in the same plane in A/159 lower left, and in different planes in A/170 where drafting conventions are more closely observed.

Fig. 7.14: Cam stack, oblique rear view.

The circular-motion cam followers (upper cam cluster) use rollers throughout, unlike the vertical motion cam followers which have sliding contact followers (vertical motion sector followers are an exception, see **Cams and Followers**, p. 170). In the case of the sector and figure wheel followers ($^{7}\mathscr{W}^{1}$, $^{7}\mathscr{W}^{2}$ and $^{6}\boldsymbol{J}^{2}$, $^{6}\boldsymbol{J}^{1}$ A/159 left) the links are driven by a simple extension of the follower arms, ($\mathscr{B}$ ^{4}F, A/159). In the case of the warning followers ($^{3}\boldsymbol{Z}^{1}$, $^{3}\boldsymbol{Z}^{2}$ A/159) the drive link ($^{4}\boldsymbol{J}$) is driven by an additional arm ($^{3}\boldsymbol{Z}^{3}$) integral with the follower arm. Unlike the vertical motion followers, where the follower 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 in the same horizontal plane (A/160). For both types of circular motion followers (i.e. with and without the additional arm) the two rollers of a follower set span the axial distance separating the two cams (A/160 shows this though identifying parts of the same assembly from this crowded drawing takes some hard peering. See rollers ^{3}F, ^{9}F associated with drive link $^{6}\boldsymbol{J}$, A/160 top right). The drive links have slotted forked ends which take pivots for the links to the rack assemblies (e.g. $^{6}\boldsymbol{J}$, A/160 top right).

The three lowermost cam pairs of the upper twelve cams in the stack generate the odd axes motions, and the uppermost three cam pairs generate the even axes motions.

The numbering of the cams in the cam stack (A/163) is problematic. Twenty-eight cams are drawn with numbering starting at $^{1}\boldsymbol{A}$ at the bottom and ending at $^{28}\boldsymbol{A}$ at the top**.** So far so good. However, there are two anomalies: the cam between $^{19}\boldsymbol{A}$ and $^{20}\boldsymbol{A}$ has no identifier. Also, for the numbering sequence from the lowermost cam ($^{1}\boldsymbol{A}$) is to be continuous then the first cam in the upper stack, identified as $^{18}\boldsymbol{A}$ (A/163), should instead be $^{17}\boldsymbol{A}$. The number sequence has been corrected below for consistency and removal of ambiguity and the new numbering has been used to identify cams in the construction drawings. The new numbers are those physically stamped on the cams.

Cam #	ID in A/163	Circular Motion Cams Description of Function
28	$^{28}\boldsymbol{A}$	Even sector drive arm – restore figure wheel valuer
27	$^{27}\boldsymbol{A}$	Even sector drive arm – return
26	$^{26}\boldsymbol{A}$	Even figure wheel drive arm – giving-off (reduce to zero)
25	$^{25}\boldsymbol{A}$	Even figure wheel drive arm – return
24	$^{24}\boldsymbol{A}$	Even warning axis – reset
23	$^{23}\boldsymbol{A}$	Even warning axis – return
22	$^{22}\boldsymbol{A}$	Odd warning axis – return
21	$^{21}\boldsymbol{A}$	Odd warning axis – reset
20		Odd figure wheel drive arm – return
19	$^{20}\boldsymbol{A}$	Odd figure wheel drive giving-off (reduce to zero)
18	$^{19}\boldsymbol{A}$	Odd sector drive arm – return
17	$^{18}\boldsymbol{A}$	Odd sector drive arm – restore figure wheel value

The spacing between circular motion cams (upper cluster) of a conjugate pair is wider than for the vertical motion cams. The figure and warning axes cams, $^{20}\boldsymbol{A}$ through $^{26}\boldsymbol{A}$, are a standard 1.25" apart; the separation of the sector axes cams, $^{18}\boldsymbol{A}$, $^{19}\boldsymbol{A}$ and $^{27}\boldsymbol{A}$, $^{28}\boldsymbol{A}$, is 0.75" (A/160, A/163). So the two outermost pairs of circular motion cams ($^{18}\boldsymbol{A}$, $^{19}\boldsymbol{A}$ and $^{27}\boldsymbol{A}$, $^{28}\boldsymbol{A}$) are more closely spaced than the four pairs 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 A/170 left where the drive arm ($^{3}\boldsymbol{Z}^{3}$) and drive link ($^{4}\boldsymbol{J}$ A/159) for the even difference warning axes is shown passing between the cams. In the case of the sector drive arms, the drive-link pivot is outside the largest cam diameter (A/170 right) which allows closer spacing.

Each figure wheel (${}_{n}{}^{2}\boldsymbol{\varDelta}^{2}$, A/171) has four decades of numerals 0-9 i.e. 40 teeth with each decade occupying one 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 (A/171). The internal drive lug of each figure wheel occupies the remaining 9° (one-digit) sector. The circular motion of the figure wheel axes consists of a fixed sweep to rotate the drive arm (${}_{n}{}^{2}\boldsymbol{E}^{2}$, A/171) from the home position to the zero-stop ($\boldsymbol{S}^{2}$) 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 drive 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 (A/171). 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 (A/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 ($^{1}\boldsymbol{R}^{2}$, A/171) refers to the maximum displacement of a sector wheel and not the sweep of the restoring arm. The 3° clearance for the restoring arm accounts for the slight difference in length of the sector and figure wheel follower arms in A/159 though the cam throws are identical. The distance from the pivot ($^{1}\mathbf{H}$) to the forked end of the figure wheel follower arm ($^{6}\boldsymbol{J}^{1}$) is 9.0". This is extended to 9.3125" for the sector follower arm ($^{7}\mathscr{W}^{2}$, A/159). This gives a 0.1" longer stroke to the sector rack which corresponds to about 3° safety clearance for the sector restoring home position.

Dispensing with Seventh Difference Carry and Warning Axes

The main elevation of the Engine shows eight figure wheel axes – seven differences axes ($^{2}\boldsymbol{\Delta}^{1}$ through $^{2}\boldsymbol{\Delta}^{7}$), and the tabular axis, $^{2}\boldsymbol{T}$ (A/163, Fig. 2.3). Plan A/164 confirms that each of the eight figure wheel axes have carry axes (consisting of helical carry arms) and warning axes i.e. there are eight carry axes shown ($^{1}\boldsymbol{C}^{0}$ through $^{1}\boldsymbol{C}^{7}$) and eight warning axes ($^{3}\mathscr{W}^{0}$ through $^{3}\mathscr{W}^{7}$ (A/163, A/164). It is clear from A/164 and A/161 that the seventh difference figure wheel axis ($^{2}\boldsymbol{\Delta}^{7}$) is provided with a carry axis ($^{1}\boldsymbol{C}^{7}$) and a warning axis ($^{3}\mathscr{W}^{7}$) in exactly the same arrangement as the other figure wheel axes. However, there is no circumstance in which the seventh difference warning and carry axes operate.

Excluding faults, the value of on any given figure wheel can only be changed by any of three distinct actions: by hand during setting initial conditions when the wheels are unlocked and free to turn; during the giving-off phase of addition from a figure wheel in a column to the immediate right (with the Engine viewed from front as in A/163); or during the carriage of tens when a carry lever increments the figure wheel following the wheel immediately below exceeding 9. The carry and warning axes play no part in the manual setting of initial values, and since there is no figure wheel axis to the right of the last figure wheel axis ($^{2}\boldsymbol{\Delta}^{7}$), the last figure wheel axis is never added to. Both the warning axis and carry axis for the seventh difference column are therefore redundant and were omitted.

With a full complement of eight warning axes, as in the original design, there would be two sets of four short rack-sections to drive circular motions for the eight warning axes – one set of four for the odd warning axes and one set of four for the evens. Dispensing with the carry mechanism for the seventh difference axis removes the need for a rack and sector gear for the seventh difference warning axis. These were omitted and the backing bar for the odd warning axes has three racks instead of the four originally specified.

Because of the introduction of mirroring of alternate columns (**Design Error**, p. 38; **3.6 Resolution of the Layout Design Error**, p. 47) the direction of the active strokes and return strokes are reversed for the odd and even axes. For example, even sector restore stroke (A/170 left) occurs when the link drives from right to left. In the mirrored arrangement, the odd sector restore stroke is from left to right. Modifications were made to achieve this.

Cam Profiles

The geometry of the cam profiles was generated working backwards from the known requirements for the circular motions. The overall shapes were then confirmed against

A/159 and A/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 ease the timing of the locking actions, and the detail of the original shapes was therefore in any event subject to minor alteration.

The even sector cams ($^{27}\boldsymbol{A}$, $^{28}\boldsymbol{A}$**,** A/170) illustrate the general relationship between cam shape and cycle timing. A/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 $^{27}\boldsymbol{A}$ at the top of the rise. This corresponds to 66° into the cycle (see 337 B 31, 337 B 314B, bottom right, for cam profile specification). 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 for the restore stroke. Cam $^{28}\boldsymbol{A}$ then becomes the active cam and drives the sectors to the zero stops so completing the stroke that restores the number given off to the figure wheels. The end of the stroke occurs 354° into the cycle (Timing Diagram 337 X 21 and 337 B 31 for cam profile). In the case of the circular motion cams the timing diagram and the angular distribution of events round the cam correspond exactly. 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 (A/170 right view, bottom centre). The dwell occupies the interval 354° to 6°. The return stroke, which occupies the interval 6° to 66° follows, during which cam $^{27}\boldsymbol{A}$ is active.

Carry Axes Intermittent Circular Motion

Main Drawings: A/165 bottom right.
Related Drawings: A/159, A/160, A/163.

The ripple-through or successive carriage of tens is performed by a helical arrangement of thirty carry arms ($^{1}\boldsymbol{C}^{1}_{1}$through $^{1}\boldsymbol{C}^{1}_{30}$, A/171) keyed to the carry shafts ($^{1}\boldsymbol{C}^{n}$) (Fig. 3.18). In the original design there are eight carry axes ($^{1}\boldsymbol{C}^{0}$ through $^{1}\boldsymbol{C}^{7}$ A/163, A/164). As described above, the carry and warning axes for the seventh difference axis are redundant and were omitted, so in the built engine there are seven carry axes not eight i.e. three odd difference carry axes and four even.

The odd and even carries occur in different half-cycles of the calculating cycle. Odd carries occur between 35 and 50 units, even carries between 10 and 25 (Timing Diagram 337 X 21). The odd difference carry axes are active during the second interval during which the even carry axes rotate but are idling: the even difference carry axes are active in the first interval while the odd carry axes idle. During the idling intervals the carry arms rotate but they are not in the same horizontal plane as the carry levers and therefore take no action: all the carry axes rotate together in the same intervals but the odd and even carry axes are active in different halves of the calculating cycle.

The intermittent circular motions of the carry axes are generated by a separate drive mechanism and not from cams in the main cam stack. The mechanism, of which gears $^{7}\mathscr{P}$ and $^{7}\mathscr{N}$ are part, is shown in A/163 alongside the right hand vertical framing member ($^{3}\mathrm{N}_{2}$), and in detail in *Fig. 4* (Plan) and *Fig. 5* (End View) in A/165. The output of the arrangement is intermittent drive to the horizontal shaft ($^{2}\boldsymbol{B}$) and the carry axes $^{1}\boldsymbol{C}^{0}$ through $^{1}\boldsymbol{C}^{6}$ are driven by bevel gears $^{2}\boldsymbol{S}^{0}$ through $^{2}\boldsymbol{S}^{6}$ pinned to the horizontal shaft (A/163).

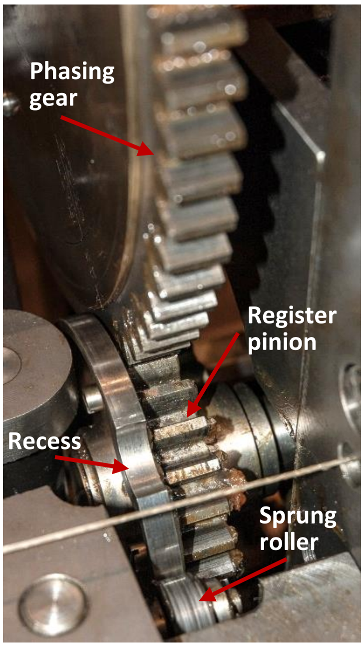


Fig. 7.15: Phasing gear.

The intermittent drive mechanism consists of a large phasing gear $^{7}\mathscr{N}$ (A/165 *Fig. 5*) with an incomplete set of teeth in two runs and keyed to the main shaft ($^{1}\mathscr{A}$), a twin-toothed gear, $^{7}\mathscr{P}$, piggy-backed on it, and a register wheel, $^{2}\boldsymbol{F}$, that drives the output shaft, $^{2}\boldsymbol{B}$, from which the carry axes derive their intermittent circular motions via bevel gears (A/163).

The twin-toothed gear, $^{7}\mathscr{P}$, is fixed proud of the plane of the phasing gear, $^{7}\mathscr{N}$, and meshes with the single impact tooth (on the register pinion), $^{2}\boldsymbol{G}$ (working points $^{2}\boldsymbol{u}$, $^{7}\boldsymbol{u}$, *Fig. 5* (End View) A/165). The register pinion, $^{2}\boldsymbol{G}$, is fixed to a register wheel, $^{2}\boldsymbol{F}$, which has a circular recess for a sprung detent roller. The sprung roller ($^{6}\mathscr{J}$), tubular spring casing ($^{3}\mathscr{S}$), and register arms ($^{4}\boldsymbol{G}_{1}$, $^{4}\boldsymbol{G}_{2}$) are shown on to the top of A/160, centre right. During the carry-axes idling periods the recess and sprung detent roller hold the register wheel 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 (337 X 21). 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 wheel to

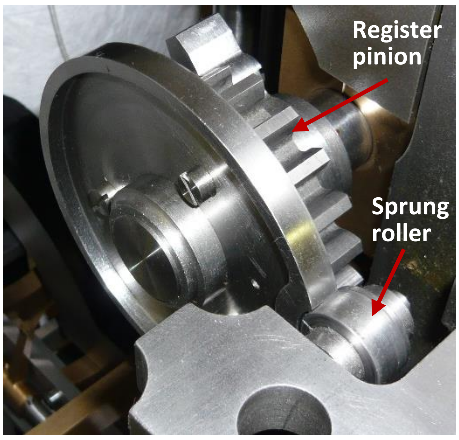


Fig. 7.16: Register pinion.

provide two episodes of carry-shaft rotation for each full calculating cycle. As described above (p. 164) the odd and even carry axes all rotate together though only one set is active during a given episode.

End View (*Fig. 5*, A/165 lower right) shows the phasing gear ($^7\mathscr{N}$) and pinion ($^2\boldsymbol{G}$) positioned at the start of a carry. The two impact teeth engage and 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 has an angular truncation to clear the trailing tooth of the phasing gear so as to leave the register pinion in its idling (stationary) position. The register pinion idles during the toothless portion of the phasing wheel's rotation and the second carry axis drive is initiated when impact teeth next mesh. Each carry occupies 104° of a 360° cycle and is followed by a 76° idling period between carries (337 X 21).

The function of the impact teeth is to provide correct meshing of the phasing gear with the register wheel when they engage. That the phasing gear and the twin-toothed gear are made as separate parts allows the twin-toothed gear to be made using 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-toothed drive is fixed to the phasing gear at 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 absorption through the spring action along the length of twin-toothed gear (A/165).

The impact tooth and twin-toothed gear are shown fixed with screws though no further details are given (*Fig. 5*, A/165). To spare the threads of the fixing screws repeated impact, the fixing holes were fitted with impact sleeves which have the action of doweling the impact tooth and the twin tooth drive to their respective host wheels. The sleeves are fixed with cheese head screws. The impact sleeves (337 D 398), were added during construction and were not included in the initial specification.

During a calculation the phasing gear $^7\mathscr{N}$ is driven continuously by the drive shaft, $^7\mathscr{V}$ (A163 and *Fig. 4*, A/165). Drive shaft $^7\mathscr{V}$ is hollow and acts as a sleeve for the main drive shaft $^1\mathscr{A}$ which passes through it (*Fig. 4*, A/165) to a journal bearing in the left framing piece (A/163). The drive train is $^7\mathscr{M}$ $^7\mathscr{V}\,^7\mathscr{N}$ $^2\boldsymbol{F}$ $^2\boldsymbol{B}$ $^2\boldsymbol{S}^n$ with Intermittency provided by the phasing gear $^7\mathscr{N}$.

In the original design (i.e. without mirroring alternate axes) the circular motion of all the

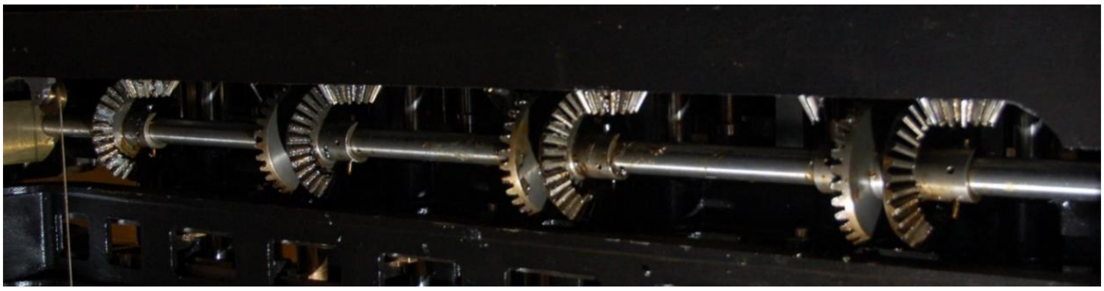


Fig. 7.17: Circular motion bevel gear drive for mirrored carry axes.

carry axes is unidirectional and anticlockwise as drawn in A/171. In the mirrored arrangement in which the odd difference axes are opposite-handed (337 X 26), the odd and even carry axes rotate in opposite directions with even carry axes rotating anticlockwise as in the original design, and odd carry axes rotating clockwise. Reversing the direction of rotation of alternate axes is accomplished by reversing the handing of alternate axes bevel-gears i.e. instead of all seven bevel gears facing the same way and evenly spaced, they are clustered in opposite-handed back-to-back pairs (Fig. 7.17).

Fig. 7.18: Back-to-back helical carry axes (rear view).

The largest single source of jams was found to be the register wheel losing synchronisation with the phasing gear when operating the Engine too fast i.e. the register wheel overrunning the detent which prevents the phasing wheel remeshing correctly when meshing is next required. The sprung roller overrunning the recess in the register wheel turned out to the determining factor in the upper limit of operable speed. Most of the jams that result from small over-throws of this kind are not damaging and are simply and quickly resolved (**User Manual (2013), Phasing Gear Jams**, p. 108-9). The detent over-throws do not always cause jams but the register pinion losing synchronisation puts the carry arms out of phase and this can cause carry lever fractures from conflict with the carry arms or, in the event of no conflict, cause carriage errors because of out of order polling.

Phasing Gear Keyway

The line through the centre of the phasing gear and the mid-point of the keyway deviates by 1° from the line bisecting the angle between the lower pair of webs (*Fig. 5*, A/165). Whether or not this is significant requires some decoding.

Working from the register wheel, the orientation of the twinned-tooth drive is determined by the geometry of the impact teeth and the correct phasing of the circular motion drive to shaft $^{2}\boldsymbol{B}$. The twin-toothed drive needs a web to fix to for support. This fixes the orientation of the webs and therefore of the phasing gear.

The phasing gear is shown keyed to $^{7}\mathscr{V}$ (*Fig. 5*, A/165) but there is no clear indication in A/163 as to how the bevel gear $^{7}\mathscr{M}$ is fixed at the crank end of the drive sleeve $^{7}\mathscr{V}$. This uncertainty is shared in A/159 (end view, bottom left in red, Fig. 7.7) where the position of the key for the clutch rocker is shown directly opposite the extended lug, but no fixing is shown for the bevel gear (A/159 plan, top centre left, in red). So it appears that the bevel and phasing gears need to be keyed because they cannot be pinned through the sleeve, that the bevel gear keyway is in line with the clutch rocker keyway (this is assumed on the basis that if the position of the bevel gear keyway was other than arbitrary, it would have been positively specified) and that, given the additional trouble of machining a keyway with an offset, the 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 keyway for the phasing gear.

A discrepancy of as little as 1° suggests the possibility of an error. It is also the case that in many of the drawings gear teeth are referenced from the flank of the tooth rather than from the centre of the tooth, or from the centre of the inter-tooth gap. The centre line through the keyway closely coincides with the line taken from the flank of the tooth opposite. This raises further doubts about how deliberate the 1° difference is. However, since the position of the phasing gear keyway, and the orientation of the phasing gear, can be determined independently there is no reason why the centre line of the keyway *should* coincide with the line bisecting the lower two webs and there is no reason in principle why the discrepancy could not have been something more substantial and thus less questionable. Had the deviation from the centre-line been greater the question of there being an error would most likely not have arisen.

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° offset of the keyway was kept to on the grounds that the offset was deliberate and dictated by timing needs.

7.4 Vertical Motions

Repeated addition required for tabulation using finite differences requires vertical motions of the locks, figure wheel, sector and warning axes (see **3. Calculation**). These motions are derived from the eight pairs of conjugate cams, $^{1}\boldsymbol{A}$ through $^{16}\boldsymbol{A}$, in the lower section of the cam stack (A/163).

Cam followers operate four pairs of horizontal bars in the section of the Engine below the eight figure wheel axes. Three pairs of bars are slung on the underside i.e. at a level just below the lower framing plate ($^{5}\mathscr{B}$, A/163) of the cam stack, and one pair ($^{24}\boldsymbol{E}$, $^{23}\boldsymbol{E}$) at a level just below the middle framing plate ($^{6}\mathscr{D}$ A/163). (Only one of the six horizontal bars of the lower set is shown in A/163 ($^{12}\boldsymbol{A}$). $^{23}\boldsymbol{E}$ is obscured by $^{24}\boldsymbol{E}$). The arrangement of cams and cam followers converts the continuous rotary motion of the cams into intermittent linear reciprocating motion of the horizontal bars. The reciprocating motion of the horizontal bars is converted to vertical motion by bell crank levers (A/163, detail A/160, A/177, Fig. 7.1). These lift and lower the axes and locks. The overall arrangement produces four sets of intermittent vertical motions for locks, figure wheel, sector and warning axes, phased according to the timing diagram (337 X 21).

Main Drawings:	A/159 (right), A/160, A/163, A/168, A/177.
Related Drawings:	A/161, A/167, A/169, 337 E.

Cams and Followers

The vertical motion cams are the eight pairs of closely spaced cams in the lower section of the cam stack. The cams were renumbered 1 through 16 with 1 the lowermost cam and 16 the topmost. In this case (unlike for the circular motion cams) the numbering corresponds to the notational identifiers for cams ${}^{1}\boldsymbol{A}$ through ${}^{16}\boldsymbol{A}$ in A/160 and by implication in A/163 (E21). Each cam is physically stamped with its identifying number.

Cam #	ID in A/159	Vertical Motion Cams Description of Function
16	${}^{16}\boldsymbol{A}$	Even warning axis lower
15	${}^{15}\boldsymbol{A}$	Even warning axis lift
14	${}^{14}\boldsymbol{A}$	Odd warning axis lower
13	${}^{13}\boldsymbol{A}$	Odd warning axis lift
12	${}^{12}\boldsymbol{A}$	Even sector axis lift
11	${}^{11}\boldsymbol{A}$	Even sector axis lower
10	${}^{10}\boldsymbol{A}$	Odd sector axis lift
9	${}^{9}\boldsymbol{A}$	Odd sector axis lower
8	${}^{8}\boldsymbol{A}$	Odd figure wheel column unlock
7	${}^{7}\boldsymbol{A}$	Odd figure wheel column lock
6	${}^{6}\boldsymbol{A}$	Even figure wheel column lock
5	${}^{5}\boldsymbol{A}$	Even figure wheel column unlock
4	${}^{4}\boldsymbol{A}$	Even figure wheel axis lift
3	${}^{3}\boldsymbol{A}$	Even figure wheel axis lower
2	${}^{2}\boldsymbol{A}$	Odd figure wheel axis lower
1	${}^{1}\boldsymbol{A}$	Odd figure wheel axis lift

Eight pairs of cam followers rock on eight separate pivots and are driven by the irregular shapes of the cams (A/159 right shows all 16 cam followers and the pivot layout; A/168 and A/169 show clearer partial views). Bar levers, integral with the cam follower bosses, slot into the horizontal bars (A/159 right). Six of the follower pivots project through the lower framing plate (${}^{5}\mathscr{B}$, A/163) of the cam stack to enable the bar levers to engage with the drive slots of the horizontal bars.

The levers operating each of the eight horizontal bars are driven by two cam followers on the same boss (A/168). Each of the cam follower arms is driven by a separate cam. Cams therefore occur in 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 drive for the vertical motions in both directions. 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 has clearance from the cam surface.

The rotation of the cams viewed from above is anticlockwise (in A/168 and physically). 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 overall layout of the vertical motion cams and followers is shown in plan view in A/159 right with additional partial views in A/168 left and right. A/159 right gives partial cam profiles with all significant rises shown in local relation to the active portion of the cam follower i.e. the cam profile of the whole cam is not shown, only illustrative sections. 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 and the rises and falls deviate inwards from this diameter.

In six of the eight cases the contact between the cam follower and the cam circumference is shown as a sliding contact (A/159 right). 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 A/159 right, and the spaghetti junction of intersecting followers with rollers in A/159 is partially clarified in A/168. A/159 right is cluttered and the multiple overlays make the trains difficult to isolate. The train for the even difference sector vertical motions is roller ${}^{9}\boldsymbol{E}$ at just past nine o'clock, ${}^{6}\boldsymbol{H}_2$ ${}^{6}\boldsymbol{K}_1$ ${}^{6}\boldsymbol{E}$ ${}^{6}\boldsymbol{K}_2$ driving horizontal bar ${}^{24}\boldsymbol{E}$. The train for the odd difference sector is ${}^{9}\boldsymbol{E}$ ${}^{5}\boldsymbol{I}_2$ ${}^{5}\boldsymbol{I}$ ${}^{5}\boldsymbol{J}_2$ driving horizontal bar ${}^{23}\boldsymbol{E}$.

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 ${}^{2}\boldsymbol{V}_{2}$, A/159 right 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 ${}^{8}\boldsymbol{N}_{1}$, A/159 lower right). It seems that Babbage was concerned to avoid point loading and so distributed the load over a contact surface. The trailing arm rises are radiused to avoid fouling: if the trailing rise were 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 3.5". 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 contactors are 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 (setting up procedures are described in **User Manual (2013)**, **Setting Initial Values,** pp. 26-35). It is possible that Babbage exploited this additional space to provide rollers where he could – 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 circular 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.

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. The bar lever ($^{7}\mathscr{K}$, A/159) driving horizontal bar $^{13}\boldsymbol{A}$ for the odd locks crosses horizontal bar $^{12}\boldsymbol{A}$ which drives the evens locks; even difference warning bar lever ($^{2}\boldsymbol{U}$) crosses horizontal bar $^{5}\boldsymbol{D}$ which drives the odd difference warnings; and even figure wheel drive lever ($^{8}\boldsymbol{M}$, A/159) crosses horizontal bar $^{12}\boldsymbol{A}$ which drives the even locks. The bars for these three motions have raised projections with slots for the bar levers as shown at the right end of the even sector bar $^{24}\boldsymbol{E}$ on A/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 ($\mathbf{E}$) is drawn ½" thick (A/160) but is labelled as 1" thick. 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 (337 E 324, E 342, E 334). This was resolved by following the vertical layout arrangement on A/163 which is drawn with the plate thickened and with sufficient clearance for the bars (only $^{24}\boldsymbol{E}$ is shown in A/163).

Specifying the Cams

Specifying the cam detail was a demanding task 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, sector wheels, 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.

Several separate features of the vertical motion cams need specifying: the outside diameter of the cam blanks from which to manufacture the cams; the height of the rises and falls to provide the required travel of the driven motions produced; the dwell to sustain the motion for the appropriate duration; the timing of each motion to produce the correct start, duration and succession of the motions for each cam of a pair and to ensure that the motions from the cam pairs are correctly phased in relation to each other.

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 in A/171 where the length of travel is inscribed inside the circles representing the shafts: vertical motion of figure wheel axis (${}^{2}\boldsymbol{\Delta}^{1}$) is given as 0.3", of the warning axis (${}^{3}\mathscr{W}^{1}$) is given as 0.34", and that of the sector axis (${}^{1}\boldsymbol{R}^{2}$) 0.68" (for full engagement with odd and even figure wheels) and 0.34" (for partial engagement while the figure wheels are restored). The significance of these figures is confirmed by the timing diagram (F/385/1; Fig. 3.2, p.21) 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 (337 E 372 A, E 371 A for example). These calculated rises were then checked against the rises shown in A/159 and A/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 (e.g. A/160). 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 that 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 in A/169 (right) and the less clear view on A/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 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 (A/159, A/169). The outer radius of the cams was taken as 6.46". This figure was calculated by triangulation and confirmed by scaling from A/159. The pitch circle radius of the cam pivots is 7.35" and the length of the cam follower arms is 3.50". 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 (${}^{6}\boldsymbol{T}_{1}$, ${}^{6}\boldsymbol{T}_{2}$, A/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 ($^6\boldsymbol{D}$) 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 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 A/159 confirms that this figure was taken as standard for all the vertical motion cams and the figure of 6.46" was used to specify the maximum outer diameter of the vertical motion cam blanks (337 E 391 C).

Design 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 for 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, a fixed pitch circle for the follower pivots, fixed follower arm lengths and tangential contact to the outer circumference, eliminates the need to cater for variant combinations of pivot position, contact angle, and arm length. These provisions significantly reduce calculation and simplify layout, manufacture and, moreover, reduce 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 trailing 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 sample calculation, **Keyway Offset – Example**, p. 190).

Cam Follower Pivots

The eight follower pivots are shown distributed around the circumference of a fixed pitch circle in A/159. Though fixed, the exact size of the pitch circle is not 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 A/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 inspected for a convenient round number. The X-Y coordinates of the even warning pivot ($^{2}\boldsymbol{D}$), for example, were measured at 4.50" and $5^{13}/_{16}$" 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.

Lifting and Lowering

The lifting and lowering action for the figure wheel axes, warning axes, and locks is provided by bell crank levers which are slotted into the horizontal bars below. The sector axes bell cranks are operated by horizontal bars, $^{23}\boldsymbol{E}$, $^{24}\boldsymbol{E}$ (A/160) from above.

The bell cranks for the figure wheel axes, warning axes and sector axes have forked ends that fit into bobbins pinned to the lower end of the axes (Figs. 7.19, 7.20, A/160 lower left, A/163). 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

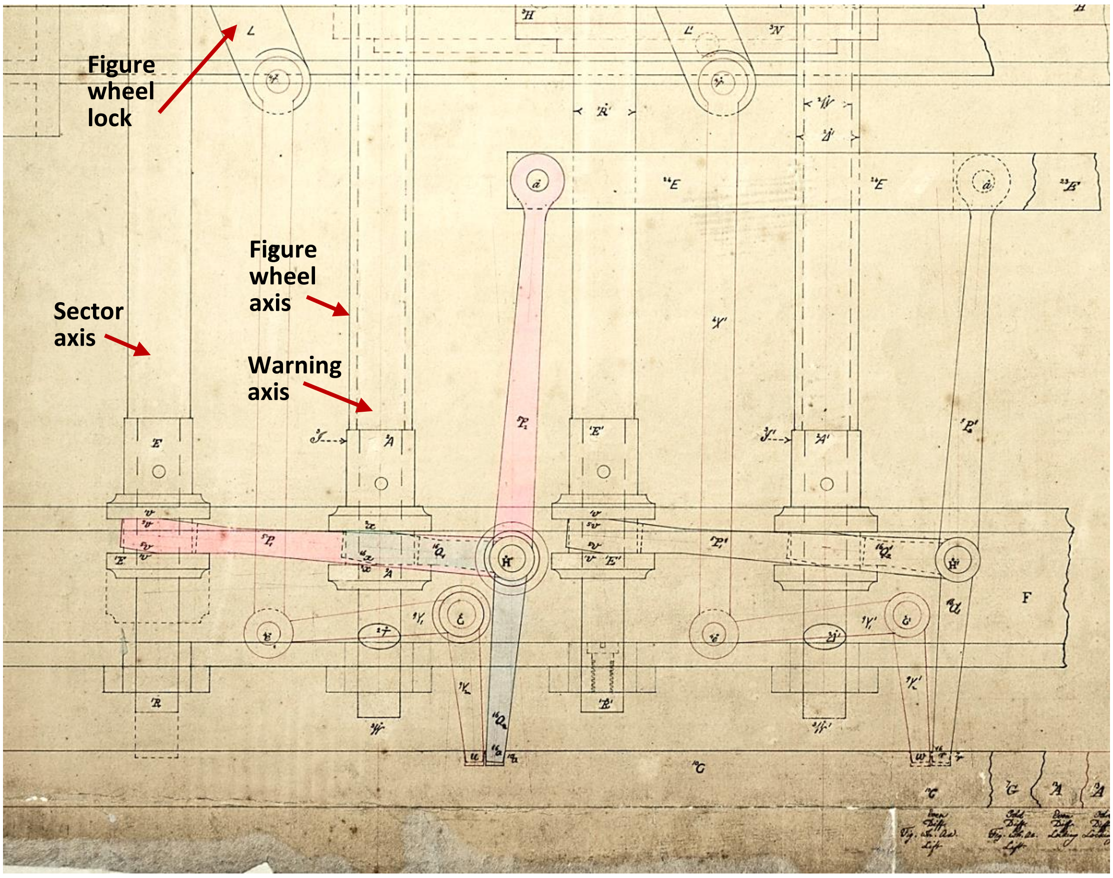


Fig. 7.19: Bell cranks for axes and figure wheel locks (A/160) (detail) (colour added).

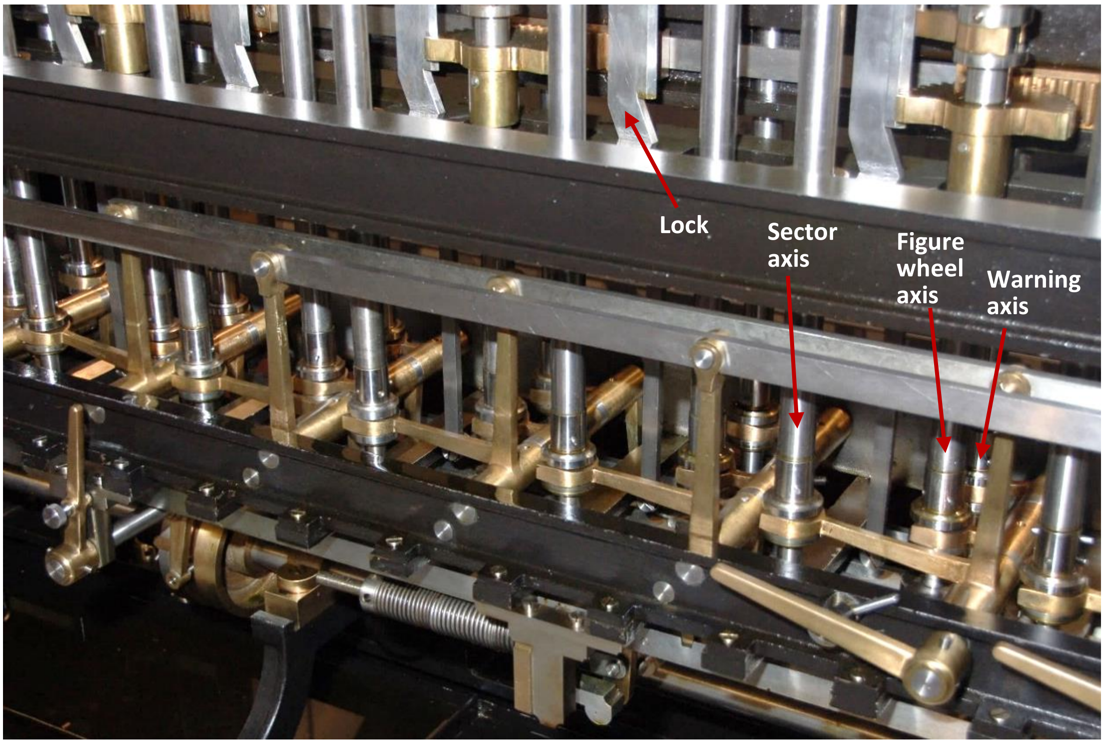


Fig. 7.20: Bell cranks for locks, sector, figure wheel, and warning axes (view from front of Engine).

bobbin and rotated in the fork by the racked drive (see **Lifting and Lowering**, p. 175).

The basic action of the bell crank and horizontal bar is clearly shown for the figure wheel axis bearing the tabular result (left-most axis A/163, Fig. 7.19, A/160 detail). The lower end of the axes project below the bobbin through a bearing hole in a plate fixed across the underside of the front and rear framing members (A/160 lower left).

The eight horizontal bars are (A/160 elevation left bottom right, A/159 plan view):

Horiz Bar	Function
${}^{10}\boldsymbol{G}$	Even difference figure wheel axis lift
${}^{7}\boldsymbol{G}$	Odd difference figure wheel axis lift
${}^{12}\boldsymbol{A}$	Even difference locks lift
${}^{13}\boldsymbol{A}$	Odd difference locks lift
${}^{4}\boldsymbol{D}$	Even difference warning axis lift
${}^{5}\boldsymbol{D}$	Odd difference warning axis lift
${}^{23}\boldsymbol{E}$	Odd difference sector axis lift
${}^{24}\boldsymbol{E}$	Even difference sector axis lift

The bell cranks (pink in Fig. 7.19) for the sector axes are operated from above (A/160) (A/163 shows the even sector bar ${}^{24}\boldsymbol{E}$ but not the bell cranks). The connection between the sector horizontal bars (${}^{23}\boldsymbol{E}$, ${}^{24}\boldsymbol{E}$) and the sector bell cranks is shown as a mix of pivots and slots: A/163 shows the even sector bar (${}^{24}\boldsymbol{E}$) with a pivot for the first and last bell cranks, and with slots for the two intermediate cranks. A/160 confirms pivots for the last even and last odd (${}^{23}\boldsymbol{E}$) crank drive. The bell crank levers for the sector lifts (two of these are shown in A/160 i.e. ${}^{5}\boldsymbol{P}_2$ and ${}^{5}\boldsymbol{P}_2{}^{1}$) share pivot shafts with the bell cranks for the figure wheel axes and warning axes. So in A/160 ${}^{5}\boldsymbol{P}_2$ (even sector drive lever) shares fixed pivot $\mathbf{H}^0$ with even figure wheel axis (${}^{2}\boldsymbol{T}$) lift driven by horizontal bar ${}^{10}\boldsymbol{G}$ and bell crank ${}^{16}\boldsymbol{Q}_2$ ${}^{16}\boldsymbol{Q}_1$ (short, blue in Fig. 7.19), as well as the with the crank for even warning axis (${}^{3}\mathscr{W}^{0}$) driven by horizontal bar ${}^{4}\boldsymbol{D}$, crank lever ${}^{7}\boldsymbol{R}_2{}^{0}$ and crank fork lever ${}^{7}\boldsymbol{R}_1{}^{0}$ acting on bobbin ${}^{3}\mathscr{I}^{0}$. (The warning axis bell crank Is not shown on A/160 but is shown in plan in A/161. The notational identifiers were taken from A/160, A/161 and from A/178/2 Notational Trains).

The layout of the bell cranks is shown in plan view in A/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 (A/160, A/177). (The warning axes bell cranks are in line with the figure wheel bell cranks front-to-back and so are obscured in Fig. 7.19, A/160. The arrangement is made clear in A/177). The bell crank shafts are located at each end in bearing holes in the horizontal framing members spanning front-to-back. 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) (A/177, Fig. 7.21). In relation to A/161 the crank pivots $\mathbf{H^0}$ through $\mathbf{H}^7$ act as shafts for three bell cranks separately driven and rotating independently on the same shaft. So In A/160 ${}^{5}\boldsymbol{P}_2$ and ${}^{16}\boldsymbol{Q}_2$ are not the same piece but are separate drive levers separately driven by horizontal bars ${}^{24}\boldsymbol{E}$ and ${}^{10}\boldsymbol{G}$ respectively. The drive levers rotate freely on pivot $\mathbf{H^0}$. They just happen to be aligned in the configuration shown in A/160.

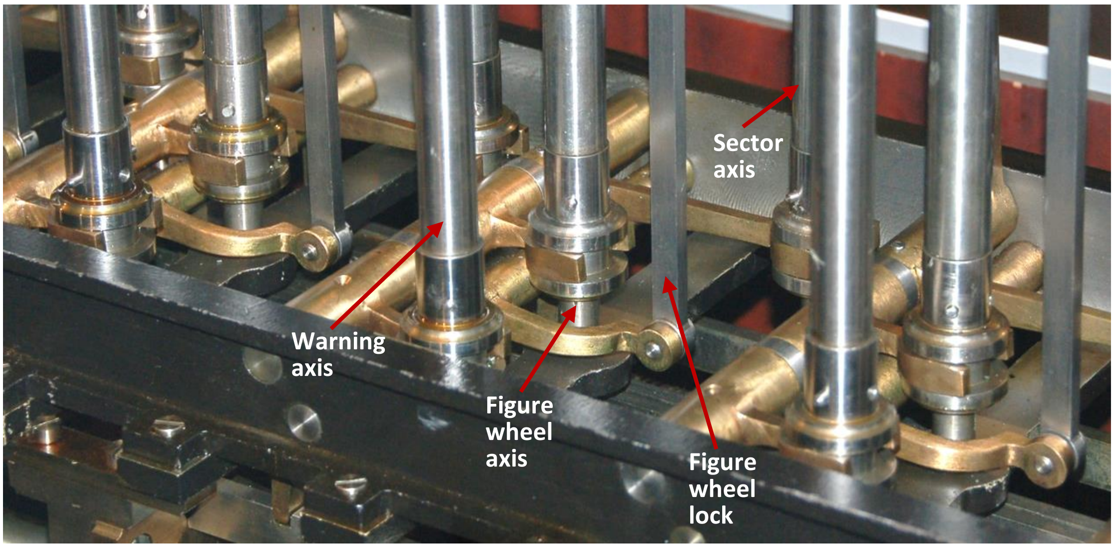


Fig. 7.21: Bell cranks for locks, sector, figure wheel, and warning axes (view from rear of Engine).

The arrangement for the locks is slightly different. Each of the lock cranks has a separate pivot shaft (Fig. 7.21, A/160, A/177). 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 (${}^{4}\boldsymbol{X}$ A/160) pivoted, at the top end, to the lower end of lock $\boldsymbol{L}$, and to the bell crank arm ${}^{9}\boldsymbol{Y}_1$ at the lower end (A/160 bottom left).

Disengaging the Sectors for Setting Up

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 by raising the sector wheels clear of engagement. The sectors are disengaged during setting up by operating the two sector lifting handles (Fig. 7.22, A/163, A/177) that lift the sectors clear of the figure wheels. Two lifting handles (${}^5\boldsymbol{E}^5$, ${}^5\boldsymbol{E}^6$ A /163) raise the sector axes to disengage the two wheel columns. The right handle disengages the even sector axes and the left handle the odd sector axes. Lifting the right handle (${}^5\boldsymbol{E}^6$) rotates a shaft to which the bell crank lever for the sixth even sectors (${}^5\boldsymbol{S}_n{}^6$) axis is pinned (A/177). This drives the even sector horizontal bar to lift the remaining even sector axes. The same process is repeated for the left handle (${}^5\boldsymbol{E}^5$) to lift the odd sector (${}^5\boldsymbol{S}_n{}^5$).

No provision is made on the original drawings for locking these in place to hold the sector axes in their raised positions so as to relieve the operator of the load during setting up. Two pull-out plungers (Fig. 7.23, F361A) were added to the front horizontal framing member, $\mathbf{F}_2$, A/163. During normal operation the plungers are locked in their home positions by bayonet locks.

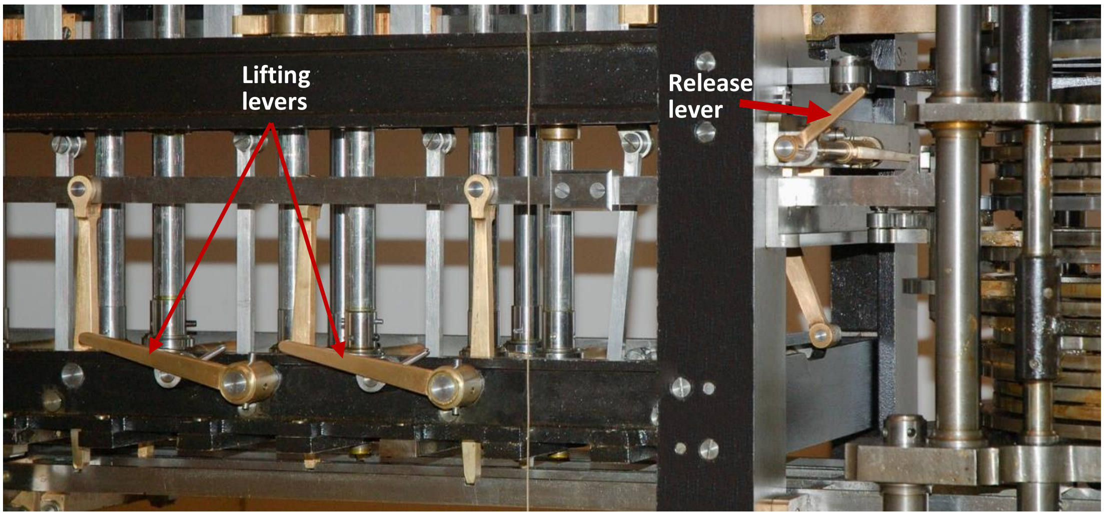


Fig. 7.22: Lifting levers and release lever (view from front of Engine).

During setting up the plungers 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.

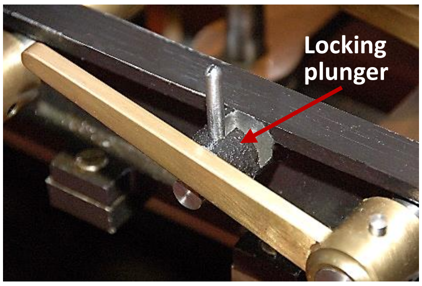


Fig. 7.23: Locking plunger.

There appears to be a dimensioning error in A/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 A/163 and on A/161. It was assumed that the shaft diameter on A/177 shown projecting from the framing member is an error. The final dimensions are shown on 337 E 394.

As well as disengaging the sector wheels from the figure wheels when setting up, it is also necessary to disengage the sector horizontal bars $^{23}\boldsymbol{E}$ and $^{24}\boldsymbol{E}$ (A/163) to uncouple the horizontal drive from the cam drive. This prevents the vertical motion drive from trying to drive the now immobilised sector horizontal bar when the Engine is advanced during the setting up cycle.

Details of the mechanism for disengagement are given in A/159 centre (in red), and A/168 top right (A/163 shows only the support bracket $\mathbf{G}$ on the cam stack side of the right upright). The follower boss, cam follower and drive lever for the horizontal bar for each of the odd and even sector bars are an integral assembly which is free to slide upwards on the cam follower pivots ($^{5}\boldsymbol{E}$, $^{6}\boldsymbol{E}$ A/168, A/159). Operating the release lever ($^{2}\mathscr{J}$) raises the two sleeves through levers $^{2}\mathscr{D}_2$ and $^{2}\mathscr{D}_1$ which lift the bar levers out of their drive slots in the horizontal bars (Fig. 7.24). Two V-notches for the release-lever spring hold the lever in the released and unreleased positions.

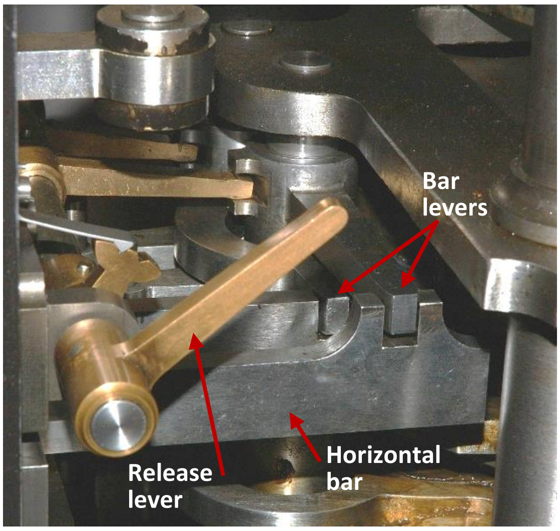


Fig. 7.24: Release lever (lowered).

Modifications

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 (right to left) 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 ($^{12}\boldsymbol{A}$ $^{13}\boldsymbol{A}$ $^{10}\boldsymbol{G}$ $^{7}\boldsymbol{G}$ $^{5}\boldsymbol{D}$, A/167) pass under the cam stack 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 printer shaft bearing mount is shown in A/167, A/163, and A/159 (faint blue pencil to the right in right view).

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 end pieces.

There is very little separation between the horizontal bars for the even figure wheel bar ($^{10}\boldsymbol{G}$ A/167) and the bar for the odd difference lock ($^{13}\boldsymbol{A}$) and A/167 shows 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 the bar supports and single broader slots provided for the two bars. Separation was instead provided by five phosphor bronze spacers fixed to and equally spaced along the length of the odd lock bar. The original separation remains unchanged. The spacers are 5/8" diameter and 0.082" thick (337 E 323).

Counterbalancing Axes and Locks

During assembly it was found that even with a 4:1 reduction in the drive the engine could not be turned past the points in the cycle where the locks are released at 25 units. The weight of the locks, friction in the angled slot-bearings, sideways pressure from the figure wheels, and the abruptness of the motion, make releasing the locks the most demanding load in the cycle. The 45° pressure angle on the lock cam (Fig. 7.25) presents a shock-load to the drive which was too great to overcome with the original arrangement.

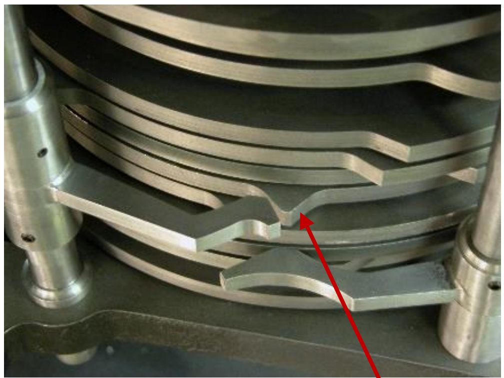


Fig. 7.25: Figure-wheel-lock cam.

Fig. 7.26: Axes counterbalancing springs.

The solution adopted was to counterbalance the weight of the locks using springs. The original design (A/163) shows a single figure wheel axis ($^{2}\boldsymbol{\Delta}^{1}$) 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 (Fig. 7.26). However, counterbalancing the locks from above in the same way is more problematic. 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 them from the figure wheels (***L***^{1}, A/160 elevation; ***L***^{0}, ***L***^{1}, ***L***^{7}, A/161 plan). 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 **Modified Seventh Difference Lock**, p. 184). The correct operation of the modified mechanism relies on the weight of the lock. If each lock was counterbalanced individually, this additional lock would need to be excepted.

These considerations 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 (337 E 22) (Fig. 6.28, p. 141).

The counterbalancing mechanism consists of a set of three springs placed end to end (Fig. 7.27). The springs are in compression and kept in axial alignment by a forked tie rod (337 E 411) threaded along its length and a threaded sleeve (337 E 419) passing through the centre of the springs. The tie rod acts on a lever which is trapped in a recessed block (337 E 414) 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.

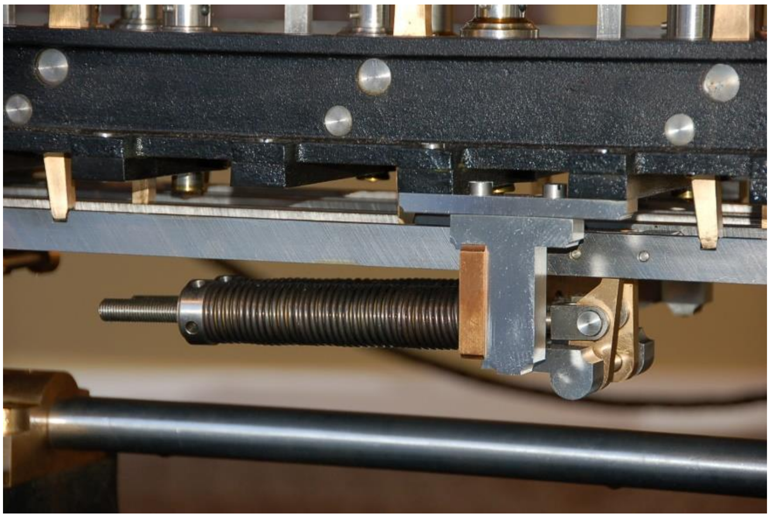


Fig. 7.27: Figure wheel locks counter-balancing springs.

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 (Fig. 7.26).

Modification to the Sector Lift Horizontal Bars

The bars ($^{23}\boldsymbol{E}$, $^{24}\boldsymbol{E}$) driving the sector bell cranks are shown supported only by the bell crank levers (A/160). The bar for the first and last even axes ($^{24}\boldsymbol{E}$ A/163) has pivot fixings for the crank levers at each end, and the two intermediate cranks are shown with slot drives. The arrangement is assumed to be repeated for the odd sector bar ($^{23}\boldsymbol{E}$) (masked in A/163 by $^{24}\boldsymbol{E}$). 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 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 in response to the loads being lifted.

The sector bars and associated bell cranks were modified. 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 (vertically) elongated holes in the bars (337 E 343, E 344). 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 i.e. the right-most figure wheel lock that is closest to the cam stack (see below 'Modified Seventh Difference Lock', p. 183). An 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 lock mechanism (337 E 22) 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 Horizontal Bar for Warning Axis Drive

The modification of the axes layout for mirroring (see **Design Error**, p. 38, p. 47) only affects the position of the carry axes, so the layout of A/177 is unaffected except for the bell crank for the seventh odd warning axis (${}^{3}\mathscr{W}^{7}$, top right A/177) which was omitted. The slot in the horizontal bar for the bell crank drive for this warning axis was also omitted.

Modified Seventh Difference Lock

The original design goes to some lengths to ensure that the figure wheels do not derange, mainly using locks to 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 and, during the carry phase, 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 and restoring. Dispensing with the seventh difference carry and warning axes as redundant leaves the seventh difference column unsecured during any interval during which it would otherwise be engaged with the sector wheels i.e. it is unsecured whenever not locked by the seventh difference lock. The seventh difference lock was modified by an additional mechanism to secure the seventh odd difference figure wheels during these unsecured intervals.

The only period in the first half cycle that the odd difference columns are secured by the odd difference locks is between 70° and 110° i.e. between giving-off odds to evens (6° to 66°) during which the odd difference figure wheels are reduced to zero, and during restoration (114° to 174°) of the number given off (Timing Diagram 337 X 21). For the other two periods in the first half cycle the odd figure wheels are secured by engagement with the odd difference sectors i.e. during giving-off and restoration. So there are no unsecured intervals during the first half cycle and no additional precautions are called for.

In the second half-cycle (evens to odds addition), odd difference columns (all of them in the original design including the seventh difference column) are secured by engagement with the even sectors during giving-off (186° to 246°) and by the carry lever horns during the carry phase (254° to 358°). However, there are no even sectors to the right of the seventh difference column and no warning axis with carry levers to secure the seventh difference column as the warning axis was dispensed with as redundant, and during these two intervals, the seventh difference column is unsecured.

The modification to the seventh difference lock secures the seventh difference column during these two intervals by locking the column for the whole of the second half cycle. This does not affect the calculation in any way as the seventh difference column retains a constant difference that is given off and restored during the first half cycle of each cycle and since, uniquely of the odd differences, the seventh difference is never added to by an eighth (even) difference, it can remain dormant through the whole of the second half cycle.

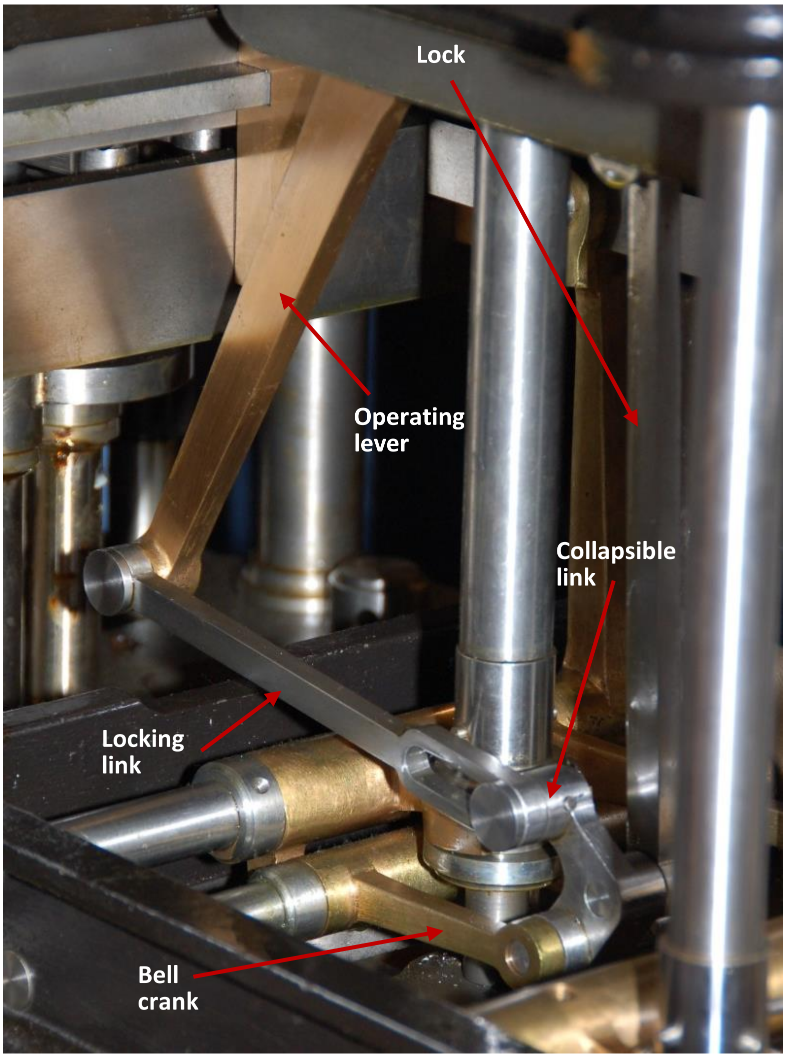


Fig. 7.28: Seventh difference lock collapsible link (view from rear).

The bell crank for the seventh odd difference lock was modified to accommodate a collapsible link inserted between the bell crank for that lock and the drive link to the seventh difference lock. The odd difference locks are driven by horizontal bar $^{13}\boldsymbol{A}$ as shown in A/160 for odd lock $^{1}\boldsymbol{L}$ driven by link $^{4}\boldsymbol{X}^{1}$ bell crank $^{9}\boldsymbol{Y}_1^{1}$ $^{9}\boldsymbol{Y}_2^{1}$ and by horizontal bar $^{13}\boldsymbol{A}$. The collapsible link is inserted at the pivot between the bell crank arm $^{9}\boldsymbol{Y}_1$ and $^{4}\boldsymbol{X}$ (the appropriate notations for this link in the seventh differed lock position are $^{9}\boldsymbol{Y}_1^{7}$ and $^{4}\boldsymbol{X}^{7}$ way off to the right, outside the limits of the drawing). The collapsible link as viewed from the back of the Engine is shown in Fig. 7.28 (the cam stack is to the left of the image). The operating lever is the diagonal bronze component and the slotted link in steel is the locking link.

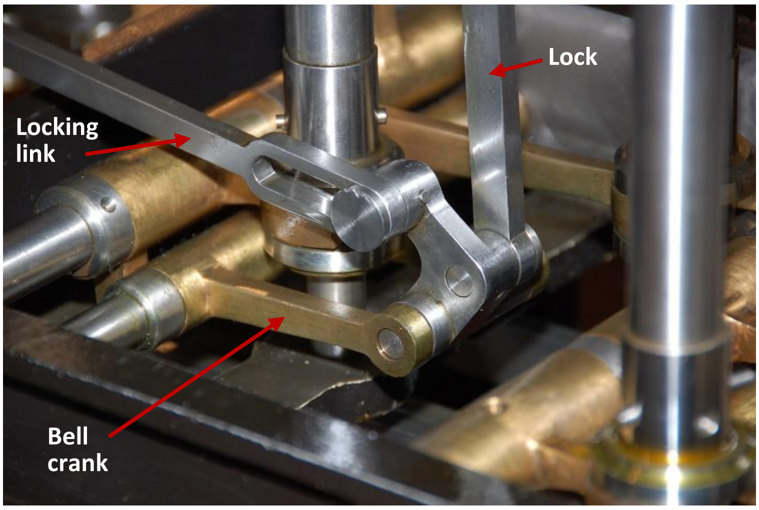


Fig. 7.29: Seventh difference lock (detail).

The seventh difference lock is driven, as are the other odd difference locks, by horizontal bar ^{13}A (A/160) which is slung underneath (not visible in Fig. 7.28). The lock is released (disengaged) when lifted and the bell crank operates, as before, by lifting the vertical drive link to the lock. However, instead of lifting the vertical link to the lock directly as for the unmodified locks the bell crank lever drives the lower pivot of the collapsible link upwards at the times in the cycle that the odd locks are lifted. Whether or not this motion is transmitted to the lock depends on the state of the collapsible link.

The link as shown in Figs. 7.28 and 7.29 is in its uncollapsed state i.e. in the position in which it transmits the crank motion to the lock to lift it into disengagement. Driving the slotted steel link (the locking link) forwards (left to right in Fig. 7.29) collapses the link i.e. hinges the link clockwise into its collapsed state and, because the pivot for the vertical drive link to the lock is offset, the collapsed link lowers the lock and in doing so engages it to immobilise the seventh difference figure wheel column. The action to lower the lock is assisted, and partly maintained, by the weight of the lock bearing downwards. So operating the slotted link collapses the link and drops the lock into engagement with the seventh difference figure wheels where it remains largely under its own weight, and with the drive between the bell crank and the lock disabled.

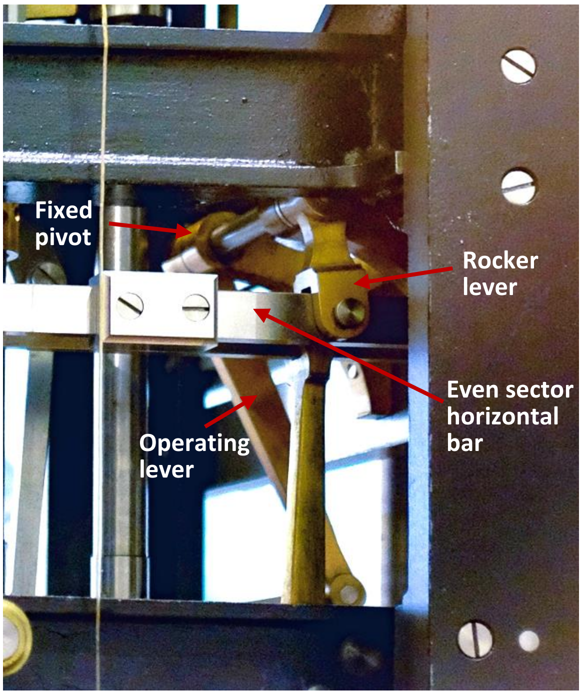


Fig. 7.30: Collapsible link drive (view from front).

The slotted link is driven by the bronze operating lever (Fig. 7.28) the drive for which is taken from the even sector horizontal bar ${}^{24}\boldsymbol{E}$ (A/163, 337 E 22). The operating lever is driven from a rocker lever pivoted on ${}^{24}\boldsymbol{E}$ via a shaft held by an additional bracket, fixed to a right-hand framing upright, to provide a fixed pivot (Fig. 7.30). The linear reciprocal motion of the even sector horizontal bar drives the operating lever of the collapsing hinge to and fro. The even sector bar ${}^{24}\boldsymbol{E}$ moves to the left (A/160 and Fig. 7.30) to lower the sectors into engagement and this occurs at the start of the second half cycle when the seventh difference lock needs to secure the seventh difference column (337 X 21). ${}^{24}\boldsymbol{E}$ moving left drives the operating lever anticlockwise (as in Fig. 7.28) driving the slotted link forward to collapse the link. With the link collapsed the brief corrective engagement of the odd difference locks that occurs at 35 units is ignored as collapsed link cannot transmit the motion and the lock is anyway already engaged.

Modification to Follower Arm Pivot Positions

A small change was made to the position of the even figure wheel axes cam follower pivot, ${}^{8}\boldsymbol{G}$ (A/159 lower right). As drawn, the cam follower boss fouls the even lock horizontal bar (${}^{12}\boldsymbol{A}$) and also fouls the end bar support (shown in blue pencil, partially 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 requires lengthening the corresponding bar lever, ${}^{8}\boldsymbol{M}$. As originally drawn there are only two different lengths for the eight bar levers. The two lower bar levers ${}^{7}\mathscr{K}$ (odd difference figure wheel lock) and ${}^{8}\boldsymbol{M}$ (even difference figure wheel axis) are one length, and the remaining six bar levers, another. To maintain only two standard lengths, 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 to ensure the correct final travel of the bars.

Timing

As described earlier the basic timing data is derived from the original timing diagram F/385/1 (Fig. 3.2) 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 more in the nature of an indicative guide rather than an exact specification. There are a number of inconsistencies in the Timing Diagram (the even difference figure wheel axis shows no return motion, for example, (though this was later corrected), and the direction of rotation for the correct operation of the warning mechanism is in error). The most serious omissions concern the timing of the locks. Unlike the motions of the calculating axes and wheels, the action of the locks is 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. The circular motion of the odd figure wheel axis, for example, is shown starting immediately the axis is lowered by 0.3" with '**L**' and '**F**' notations alongside the arrows indicating locked and unlocked states. 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 the Timing Diagram (F/385/1) shows the wheels 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 is a feature of the diagram. Inexactly specified timing intervals are not confined to locking action.

The new timing diagram (337 X 21) supplements the original by providing the detailed lapping and phasing information otherwise lacking. 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 retained for convenience of reference but was not used in the modern specification.

Lapping and Clearance

In the case of lifts from leading followers, the rise in the cam profile starts at the point in the cycle at which the motion starts i.e. the point of contact with the follower coincides with the start of the rise, and 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 $^{1}\boldsymbol{A}$ lifts the axis and $^{2}\boldsymbol{A}$ lowers it. The lifting action on the cam occurs between 67° and 75° which gives the start of the leading rise, and the duration. The corresponding fall on $^{2}\boldsymbol{A}$ is given a few degrees clearance and is specified between 64° to 73° (337 E 373 A). The few degrees timing clearance correspond to a clearance of about 0.003" between the passive follower and $^{2}\boldsymbol{A}$. The action to lower the axis is determined by the rise on $^{2}\boldsymbol{A}$. This action occurs between 357° and 5° and a corresponding clearance is allowed on the fall on $^{1}\boldsymbol{A}$ (337 E 374 A). 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 (337 X 21) shows the lifting action starting 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 that is active on the plateaus. A 1° lag in the cam rise is introduced to compensate.

Allowance was made for the roller followers used for the sector vertical motions. 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 of the Locks

The cams for the locks presented special difficulties. A single calculating cycle requires four separate episodes of engagement of the figure wheel locks, at 10, 25, 35 and 50 units (337 X 21). Three of these are short corrective engagements to re-align minor derangements and to secure the figure wheels during momentary otherwise unsecured intervals. These occur at 10, 25, and 50 units for even figure wheel locks, and at 25, 35, 50 units for odd figure wheel locks. The longer locking periods (at 35 units for evens, and 10 for odds, 337 X 21) are 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 restore their 'lost' value.

In instances where Babbage made allowances for the time taken for entry and withdrawal locks he allocated a single nominal timing unit corresponding to 7.2° in a 360° cycle (F/385/1). However, working back from the rises on the locking cams shown in A/169 the actual period for locking and unlocking occupies 10° (4° for entry, 2° for dwell, and 4° for withdrawal). It is clear from the cam details in A/169 and A/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. In the modern implementation the minimum interval is 11° (4° plus 3° plus 4°) primarily to ease the pressure angle of the rise. The difficulty was then to fit these slightly stretched locking actions (particularly the three short engagements) into the already 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 of the figure wheels that follows. The dwell of the locks was extended slightly in some instances to cater for the exact locus of the point of contact of the follower on the cam profile.

Phasing the Vertical Motions

The cams are keyed to the central cam drive shaft (cam $^{1}\boldsymbol{A}$ is the sole exception, see below **Driving the Cams**, p. 192). The position of the keys is critical 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°.

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 sixteen cams needs to be calculated separately. The keyway calculation for the even warning cams (cams 15 and 16) is described in the following example.

Keyway Offset — Example

The first step is to calculate the angular displacement (Θ) of the pivot centre. The coordinates of the even warning pivot are given by the pitch circle radius (7.35") and the distance from the horizontal axis (4.50" scaled from A/159). 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". 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 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 of 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" line 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 is 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 a round and convenient 56°. This translates into a clockwise displacement of keyway for the trailing cam (Cam 15) of 260°15′.

Driving the Cams

Fig. 7.31: Three paired vertical-motion cams.

All the vertical motion cams were made from identical blanks (337 E 391 C) 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 (Cams 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. A/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 (^{1}***A***). The lower cam shaft bearing prevents Cam 1 being keyed to the shaft. The boss on ^{1}***A*** was removed to clear the cam shaft bearing and Cam 1 is screwed and dowelled to Cam 2. 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 keyed and each cam is screwed and dowelled to its mating cam. Cam 1 is the only cam not keyed to the shaft. Fig. 7.31 shows three pairs of vertical motion cams screwed and dowelled in pairs with bosses removed. (The cam at bottom right is a circular motion cam.)

Verification

The cam rises and phasing were verified graphically. Tracings of the cams were overlaid on the mating cam 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 cut-outs. (See 337 E 383 A&B for warning cams (leading); 337 E 384 A&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.

For discussion of the symmetry of rises and falls in conjugate cams, the direct generation of cam profiles using inked rollers, subsequent manufacture and verification, see **Chapter 9**, **Build, Cam Profiles**, pp. 202-4.

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 (337 E 353&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 marked before being slid off the pivot shaft, were welded together, slid back on and then pinned to the pivot shaft. There was an awareness that this procedure might 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 before adjustment and welding to ensure that any heat distortion from the hardening process did not affect the trial settings.

8. Framing

Main Drawings: A/161, A/163, A/164, A/176, 337 F, 337 A.

'Framing ' refers to the fixed structure supporting the working parts of the Engine. It includes the two base support rails 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, drive shaft mountings, and cam stack framing.

8.1 Calculating Section

The calculating axes are contained between bearing plates which span, front to rear, the upper and lower horizontal frames or trays (A/163, A/164). The upper frame ($\mathbf{O}_1$, $\mathbf{O}_2$, $\mathbf{O}_3$, $\mathbf{O}_8$ A/164) is shown resting on uprights (${}^1\mathbf{B}_1$, ${}^2\mathbf{B}_1$ (rear); ${}^4\mathbf{B}_2$ ${}^3\mathbf{B}_2$ (front) A/163, A/176) and the lower frame (${}^3\mathbf{N}_1$ ${}^3\mathbf{N}_2$ ${}^3\mathbf{N}_3$ ${}^3\mathbf{N}_4$, A/176 roughly midway up the engine) is shown with open corners let in to the uprights (A/161, A/176). The upper frame is shown as a single tray (A/164), possibly as a one-piece casting or, given the absence of piece-part detail, the single-tray representation is possibly a drafting shorthand. There is no indication on the elevation (A/163) of piece part assembly of the frame. Similarly, the sub-assembly of the lower tray is unclear (A/161, A/176). No fixing is shown for the upper (A/163) or lower frames (A/161, A/176) to the uprights. (The fixing on the top left of A/161 is for the rack support).

There is an inconsistency in the dimensions given for the overall length of the frame as annotated at the bottom of A/161 ('*5 feet 2 ¼ inches*') 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 A/171 is 6.98". Seven such pitches give the distance between the first and last figure wheel axes as 48.86". Adding to this the scaled dimensions to each end of the frame (5" to the right and 8.14" to the left) gives a figure of 62" for the overall length of the frame. The annotation at the bottom of A/161 gives 62¼ " for this same overall dimension – a discrepancy of 0.25". The scaled figure of 62" was used and the annotation ignored.

Because of the difficulty casting large one-piece framing pieces, and the difficulty machining the surfaces for the bearing plates inside a tray, 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 337 F 311 A-D), and a pair of members (long dimension, front and rear 337 F 312 A-D). In the notations for the framing pieces, each of the four framing members for each of the two trays have the same part identifier letter, the same Index of Identity, and different indices of Linear Position (Fig. 1.2). The Notation therefore suggests that the trays were intended to be made in four sections though the use of the notations in the Engine drawings overall does not always adhere to the declared rules for lettering drawings.

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 (long) member of the lower tray (A/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 differently machined. The upper and lower frame ends have different vertical depths. The lower right-hand frame end (337 F 311 B) takes the main drive bearing, and loose pieces are added to the pattern to provide for this. The top left frame end (337 F 311 D) has mounting holes for the printing mechanism, a central circular clearance cut-out, and a rectangular cut-out. The need for clearance cut-outs 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.

Details of the cross-bracing arrangement and piece-part drawings are in the A-series construction drawings. Procedures for bracing the Engine for transport are described in the **User Manual (2013)**, **Moving the Engine,** p. 127.

Front and Rear Framing Pieces

The top and bottom bearing plates (for examples of lower bearing plates see A/176) span the front and rear framing members and are fixed to a raised machined lip along the top of the inside bottom flange of the angle (Tracing BAB/B/006). The height of this machined surface was taken as 1/8" (337 F 312 C&D). This dimension is taken from the depth of the internal pad on the upright intended to receive the lower frame (A/176). Pads for pulley brackets (for the gut cord to operate the uncoupling clutch A/163) are required on the right-hand side only, front and rear framing pieces. (The rear pulley is an additional one fitted to counterbalance the scoop cam clutch for end-of-page halting (Fig. 7.10 .p. 156). Only these two pads were machined. Redundant pads on left front and rear are unsighted on the inside and were left intact after casting.

The four long framing pieces were similarly cast from one pattern, differently machined. Differences between upper and lower members include: vertical depth; the omission of curves on the underside of the upper framing pieces (337 F 31 A - D, A/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, and 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, this view does not give full weight to the structural role of the figure wheel supports (A/176) 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 (Figs. 3.6, 3.14). 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 not pursued and the original dimensions were adhered to.

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 two long frame members screw to the frame ends and are lapped. Fixing is by tapped recessed cheese head screws.

Rack Mounting

Main Drawings: A/160, A/161, A/163, A/171

Sets of toothed racks meshing with gear sectors provide the intermittent reciprocating circular motion for the odd and even figure wheel axes, the odd and even sector axes and the odd and even warning axes (Figs. 7.11, 7.12, A/171 bottom right). The racks are mounted on backing bars and run in channels machined in the rack mountings (see **7.3 Circular Motions**, p. 157).

The layout of the circular motion rack mountings for the figure wheel axes are well defined in A/163. The same cannot be said for the rack layout and sector gears for warning and sector axes the layout for which is not immediately obvious from the density of A/160. The racks for the circular motions of the four odd figure wheel axes are mounted in a single machined casting ($^{6}\boldsymbol{I}$, A/160), and similarly for the four even figure wheel axes with the single machined casting $^{9}\boldsymbol{I}$. The rack mounting for the odd warning axes is also one machined casting ($^{4}\boldsymbol{K}$) and similarly for the even warning axes, $^{5}\boldsymbol{K}$.

The figure wheel axes and the warning axes are in line so the first odd warning axis (seventh difference, extreme right) is masked in A/163. However, A/163 shows the bevel drive for the first odd carry shaft indicating the inclusion of a carry mechanism here. The original intention to include a full carry mechanism for the seventh difference is confirmed in A/161 and A/164. The seventh difference carry and warning axes were omitted as redundant as described earlier (p. 162). The profile of the curve for the right-hand end of the figure wheel rack mounting (337 F 35) was taken from $^{2}\mathbf{J}$ (A/163).

The figure wheel rack mounting ($^{2}\mathbf{J}$, A/163) is shown supported at the base only (A/163, A/160 bottom left). Three additional mounting ears were added to the top of the casting to provide additional back-to-front support (337 F 351). 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

Bearing plates, supporting the axes top and bottom, span the front and rear members of the framing members of the upper and lower trays (A/176, Fig. 8.1). In both trays the plates rest on machined lips running the full length of the trays (337 F 312). The framing plates support figure wheel axes, sector wheel axes, carry and warning axes, locks, twenty-four figure wheel supports, sliders for the zero stop pillars (three for each figure wheel axis, **O**, **P**, **Q**, A/176), and sector wheel zero stop pillars. The length of the figure wheel support pillars is finely machined and they serve to both strengthen the frame and share the load of the axes between the upper and lower trays.

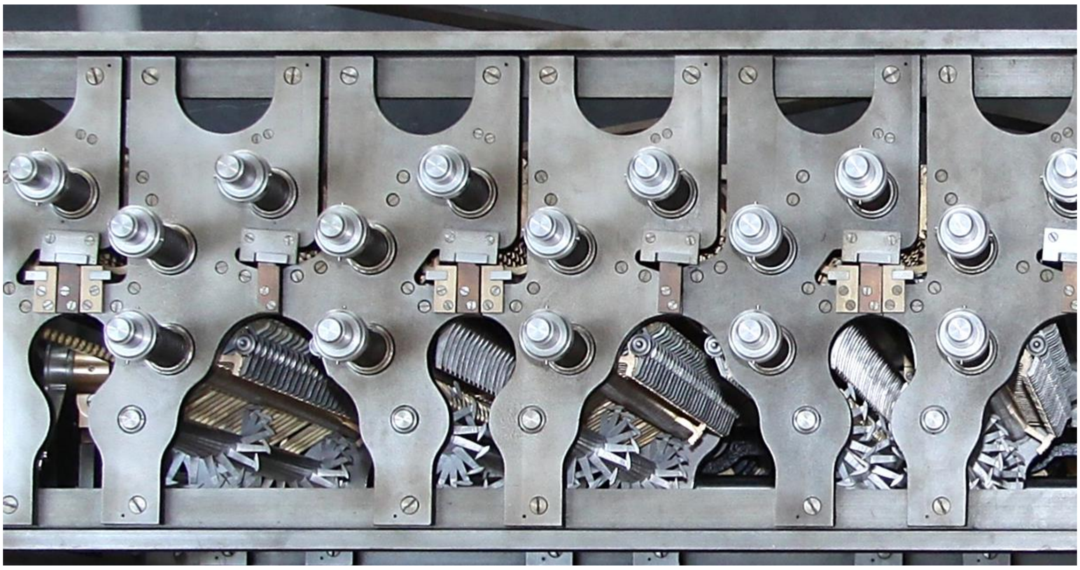


Fig. 8.1: Upper framing plates. View from above.

The relative positions of the addition and carry mechanism components are shown in A/164, A/171 and A/176. In each of these three drawings both odd and even carry mechanisms are shown handed to the right. The mirroring correction of alternate axes (p. 47) requires the odd numbered axes to be opposite handed (Fig. 3.22, 337 X 26) i.e. with the carry axes handed to the left, the position of the locks, figure wheel supports, carry axis and sector wheel zero stops invited review. The odd and even bearing plates, drawn in the original as identical ceased to be so. The positions of the locks are unchanged but there are other modifications in consequence of mirroring.

The differences between upper and lower bearing plates are small but significant. 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 (fixing holes are not shown in the original drawings). 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 (seventh odd difference column) has the warning and carry-axes shaft bearing holes omitted as the carry mechanism for this axis was omitted, and the position of fixing holes were altered to accommodate the new back-to-back placement of the figure wheel zero stops (compare mirrored layout in 337 X 26 with A/171). Bearing plate variants are shown in construction drawing 337 F 37.

Base Supports and Shaft Mountings

Main Drawings: A/159, A/163, A/165, A/167, 337 F 381 A&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. A/163 shows the right-hand vertical framing piece (${}^{3}\mathbf{B}_2$) resting on a boxed platform cast into the base support, and the left vertical (${}^{4}\mathbf{B}_2$) resting on a cantilevered platform. The same is assumed to be the case for the rear two uprights (${}^{1}\mathbf{B}_1$ (left), and ${}^{2}\mathbf{B}_1$ (right)). The left-hand cam stack pillar (${}^{1}\mathscr{A}_1$) is also shown resting on a cantilevered platform (A/163 and A/167). The right cam stack pillars rest on cylindrical bosses on the end mounts (A/167, 337 F 383).

The three drawings A/163, A/165, A/167 show three shaft mountings along the length of the output apparatus drive shaft (${}^{4}\mathscr{C}$). The shaft mountings span the two base supports. One shaft mounting is provided at each end of the base supports, **D** (left), **B** (right), with the third ($\mathbf{C}$) in between to the left of midpoint. A fourth shaft mounting was provided in addition just inside left of the printer drive gear for additional support. The three left-most shaft mountings are identical (337 F 382). The right-hand end mounting (337 F 383) has two bosses for the cam stack pillars and two half-bossed holes for the sliding bars support (A/167 left).

Each of the base support rails ($\mathbf{A}_1$, A/163 is thickened in four places by the addition of pads on the lower surface of the base supports positioned under the verticals – this to cover the protrusion of the cam stack pillar bolt (A/167 bottom left) and to provide a machined surface for levelling jacks.

Cam Stack Support

A/167 shows the cam stack pillars secured below to the base supports. No upper fixing is shown and the drive shaft assembly from the crank provides the only other fixing for the cam stack frame. Two Upper framing ties (Fig. 8.2) were added to keep the drive shaft aligned and to reduce the risk of the machine breaking its back when moved. A/164 and A/159 show plan views of the cam stack framing plates. The upper framing plate (337 F 424) was extended by the addition of two ears similar to the those shown for the lower framing plate (A/159). The upper framing ties (337 F 423 A&B) fix the ears to the verticals to secure the upper portion of the cam stack.

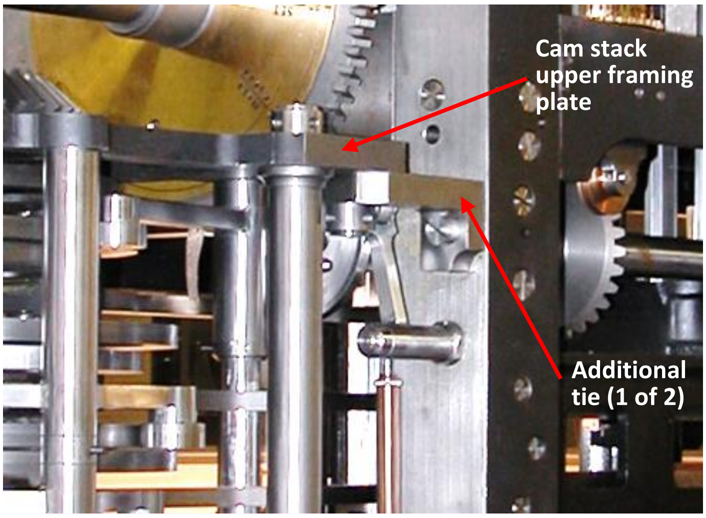


Fig. 8.2: Additional ties for cam stack frame.

Drive Shaft Bearing

There is an inconsistency in the dimensions given for the drive shaft bearing ($^5\mathscr{F}$, $^6\mathscr{F}$, A/163 on top of the cam stack) as shown in A/159 and A/163. In the full-sized drawing of the bearing in A/159 (faint blue lower left between the two notations tables) the dimension from the bottom of the base of the bearing to the shaft ($^1\mathscr{A}$) centre is drawn as 5¼". A/163 shows same dimension at 4½". The discrepancy is accounted for by the difference in the drawn thicknesses of the cam stack framing plates in A/160 and the thickness dimensions annotated in A/159. Specifically, the bottom cam stack framing plate is drawn as ½" thick in A/160 though the manuscript annotation in A/159 (top left corner of left-hand drawing) gives the thickness as 1". Similarly, the upper framing plate is drawn as ½" thick in A/160 but the handwritten annotation in A/159 calls for ¾" (top right corner of left-hand view). The middle framing plate is drawn in A/160 ½" thick and annotated in A/159 as ½". The discrepancy between the plate thicknesses as annotated and the thicknesses as drawn is ¾" which is the difference between the distances to the shaft centre as shown variously in A/159 and A/163. The annotated thicknesses were used rather than the drawn thicknesses on the basis that these superseded the already-drawn plate thicknesses in A/160 and the end view of the bearing in A/159, and the distance to the shaft centre was implemented as 4½" (337 F 421). The plate thicknesses appear to have been amended by annotation on A/159 without amending the shaft bearing drawing in A/159, or the cam stack layout (A/160) which had already been drawn. A/163 was clearly drawn to the annotated dimensions and was used elsewhere in the original drawings in the layout of the machine.

8.2 Output Apparatus

The framing and modifications to the framing for the printing and stereotyping apparatus are described in **Additional Modifications, 8. Modifications to Frame**, p. 146-7.

9. Build

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

Base Rails

The Engine rests on two cast base rails that run the full length of the machine front and rear (A/163). Cast base rails were delivered dog-legged at the printer end i.e. splayed outwards. Time pressure to complete was against having them remade despite contractual entitlement to replacements. Fixing-holes for the upright framing members and for the legs supporting the printer drive shaft bearings (which span the rails) had been drilled in the machined palms on the rails but were not aligned to correct for the distortion. It appears that they were marked out by reference to two inconsistent datums possibly resulting from two separate stages of planing. The issue was solved by elongating the fixing holes in the shaft bearing supports and in the base of the uprights and using the supplied rails modified in this way.

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 (A/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 are at least partly self-cancelling.

In the example cited, the rise of the main cam is taken as the drive. In the general case it cannot be assumed that all rises, whether on the main cam or 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 biased from the neutral position by a counterbalancing spring. 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 rises and falls in the profiles are proportionately modest. In contrast, the circular motions cams show more extreme variation. So the effect of the non-ideal behaviour of the followers was 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 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 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.

The start and stop positions of the rises and falls are critical to timing. 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. the 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 angles 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 window 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 freed up using a run-in speed of 10 complete engine cycles per minute i.e. one tabular calculation every six seconds (40 turns of the handle per minute with the 4:1 reduction gear). If the machine is run faster than this, the sprung roller of the intermittent circular motion drive of the carry axes tends to overshoot its recess in the register pinion and cause jamming of the drive (p. 167). If the motion of the main crank is erratic or too slow, 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 and, at slow speeds, producing timing lags. With sufficient uniform momentum the meshing is unproblematic in normal operation.

Cam Keyways

Since the lead-in and lead-out timings are critical and since 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.

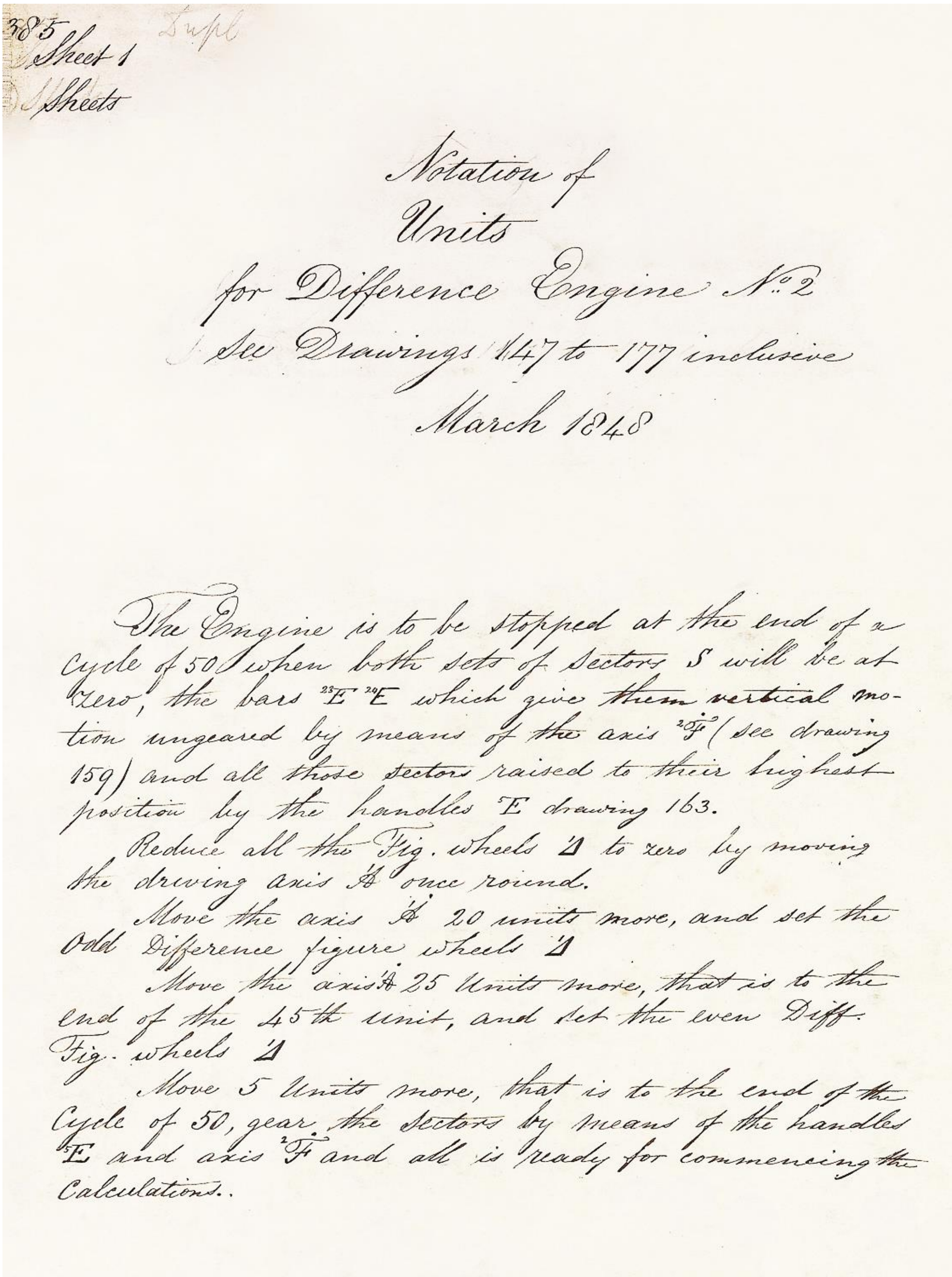
Figure Wheel Drive Arms

A Trial Piece (Fig. 3.14, p. 30) was built to verify the new layout in which alternate axes are mirrored. The internal drive arms fixed to the figure wheel drive axes 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 into the barrels of the figure wheels themselves. On the Trial Piece the drive arms did not foul the internal nibs and there was no advance indication that fouling might be a problem. However, with multiple figure wheel stacks, 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.

Setting Up Initial Values – Manual Figure Wheel Locks

The setting up procedure consists of a series of operations that allow the figure wheels to be set manually to the initial values of differences. This includes setting the tabular value from which tabulation is to start. 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 horns 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 (p. 35). These measures are intended to prevent derangement of the wheels during normal operation i.e. movement imparted by extraneous action, deliberate or inadvertent. The security measures prevent inputting initial values by manually altering figure wheel settings at an arbitrary time and these security devices need to be disabled or bypassed to allow the figure wheels to be turned by hand. An exact procedure is required to set up initial values.

Babbage provides no systematic description or explanatory text for the Engine as a whole. In a rare exception a brief textual account is given (F/385/1, Fig. 9.1) for the setting up procedure as a preface to the timing diagram (F/385/1) both 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. Initial values are entered during the second full cycle. The procedure described by Babbage has a major flaw which became evident when the procedure was attempted in practice.



385 Sheet 1 Dupl
Sheets

Notation of
Units
for Difference Engine No. 2
See 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 Δ to zero by moving the driving axis A once round.

Move the axis A 20 units more, and set the Odd Difference figure wheels Δ

Move the axis A 25 Units more, that is to the end of the 45th unit, and set the even Diff. Fig. wheels Δ

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.

Fig. 9.1: Original setting up procedure for initial values (F/385/1, March 1848).

The italicised sections below are transcriptions of the original text:

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

The first instruction is to disengage the sector wheels from the figure wheels. The timing diagram (which follows the manuscript description confirms that both odd and even sectors are at zero at the end of the second half-cycle. Disengaging the sectors from the figure wheels by raising them allows the figure wheels to be turned by hand without giving-off to the adjacent lower difference column while the initial values are entered and, when the cycle is advanced, during the rest of the setting up procedure. Both sets of sector wheels are fully disengaged (raised to their highest point) for the duration of the setting up procedure.

The sectors are disengaged by operating the two lifting handles (Figs. 7.22, 7.23, $^{5}E^{5}$ $^{5}E^{6}$ A/163, detail A/177). The left-hand handle lifts the odd sector axes to the fully raised position; the right-hand handle lifts the even sector axes. At the end of the second half-cycle ('*the end of a cycle of 50*') the odd sectors are 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 levers at the cam stack end. This is achieved by lifting the release lever ($^{2}\mathscr{F}$, A/159 lower centre in red between the two notation tables, with clear detail in A/168 top right viewed in portrait mode) which lifts both the odd and even bar levers out of their drive slots in the sector bars (Fig. 7.24). This frees the lifting handle to move the sector horizontal bar. The lifting handle is locked in the raised position, as before, with the pull-out plunger (Fig. 7.23). Disengaging the sector bars prevents the vertical motion drive from conflicting with the now immobilised sector axes when the engine is cycled during the setting up procedure. (The lifting handles waggle slightly when the Engine is run).

> *Reduce all the Fig. wheels* $^{1}\Delta$ *to zero by moving the driving axis* $^{1}\mathscr{A}$ *once round.*

This step contains a major flaw which allows the setting up process being self-corrupting. The timing diagram shows that at the start of the cycle the odd figure wheel locks disengage, the odd figure wheel axes lower to bring the drive arms into the plane of the internal nibs, and the odd figure wheels are then reduced 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 driven by the drive arms, or are locked, or are being restored by the sectors i.e. during normal operation the figure wheel motions are defined and systematically constrained throughout. However, the first step of the setting 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, the figure wheels are not secured and it was found that during the return of the figure wheel axes, friction between the axis and the figure wheels dragged back some of the figure wheels a varying amount depending on the level of frictional drag, and displaced them from zero by an indeterminate amount.

The occurrence of an interval during which the figure wheels are unsecured subverts the intended zeroising action. 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 load on the figure wheel would be well in excess on anything that slight friction could derange. 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 this self-corrupting process. The amount of frictional derangement is indeterminate and any derangement that is not an integral angular displacement of one digit will foul the locks at next entry. Even if jamming the locks was not an issue, the same unsecured dragging action of the figure wheel axes would corrupt the initial values during the second cycle of the setting up procedure.

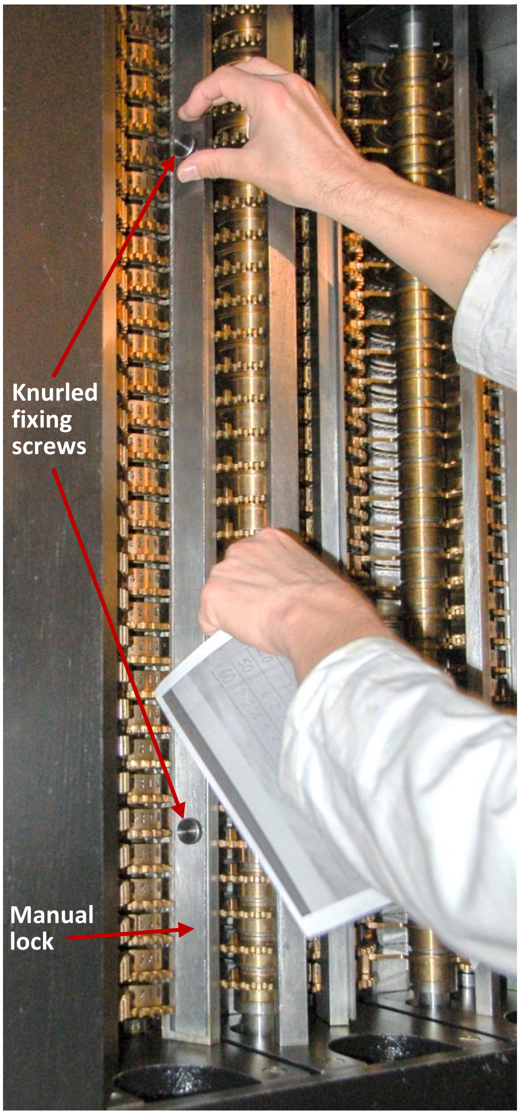


Fig. 9.2: Operating manual locks.

The solution was to provide manual locks for each of the axes. These consist of vertical slats 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 thumb 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 locks, which run the full length of the figure wheel columns, when slid forwards, insert between the teeth of all the figure wheels in the full-height column, and immobilise them in exactly the same way as do the vertical locks activated automatically by the drive mechanisms. When engaged with the column of figure wheel teeth the locks are 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 values 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.

> *Move the axis $^{1}\mathscr{A}$ 20 units more, and set the Odd Difference Figure wheels $^{1}\boldsymbol{\Delta}$.*

This is the start of the second full cycle of the setting up procedure. The 20-unit point occurs during the period of normal operation in which the odd 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 and the odd difference figure wheels can be turned in either direction without producing spurious warnings i.e. wheels can be turned to any of the four decades without transitions between 9 and 0 setting carry warnings. Warnings set during the setting up procedure before the 17-unit point are cleared automatically; those set after the 20-unit point would remain as residual warnings and have to be cleared by hand. The advantage of setting odd difference initial values at the 20-unit point is twofold: there are no spurious odd 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.

> *Move the axis $^{1}\mathscr{A}$ 25 units more, that is to the end of the 45th unit, and set the even Diff. Fig. wheels $^{1}\boldsymbol{\Delta}$.*

The 45-unit point is the even-axis equivalent of the 20-unit point for the odds axis and the same considerations that apply to the odd differences initial value settings apply.

> *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 $^{2}\mathscr{F}$ and all is ready for commencing the calculations.*

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

Revised Setup Procedure

The setting up procedure was revised to include the use of manual locks to solve the problem of self-corruption in Babbage's original sequence. The need to adhere strictly to a fixed stepwise setting up procedure cannot be overstated. Practically all instances of damage to the machine arose through inattention during setting up, missing specific steps in the procedures, or deliberately introducing short-cuts that have proved to have unintended consequences.

The bronze carry levers are particularly vulnerable. During the carry phase of the addition cycle steel carry arms rotate to service warned figure wheel positions. If a warning is set, the locus of the steel arm intersects with the position of the bronze carry lever. The action of the steel arm is to sweep the carry lever aside and in so doing advance the next figure wheel up by one digit. If the figure wheels are locked, as they are by the manual locks during the setting up procedure, the bronze carry levers, engaged with the figure wheels by the carry lever horns, cannot move and the carry arms foul the carry levers. The forces required for the steel carry arms to snap the bronze carry levers is not sufficient to cause a jam and the carry levers fracture. The back-pressure felt in the crank handle is barely noticeable especially in view of the 4:1 reduction gear in the crank. Continuing to drive the Engine, especially by operators unused to the feel of the crank during normal operation, can readily strip a warning axis of carry levers.

It is therefore imperative to adhere to each step of the setting up procedure especially as the procedure relates to the use of the manual locks which have to be withdrawn before a tabulation is started.

Babbage's original manuscript description of the setting up procedures calls for two complete calculation cycles. The first leaves all figure wheels zeroised, and during the second cycle odd and even difference initial values are entered. Zeroising figure wheels provides the opportunity for a convenient all-zero visual check of the correct operation of giving-off and carry reset, but apart from this there is no obvious reason why the first cycle cannot be omitted.

Both the two-cycle and single-cycle setting up procedures were used, and the single-cycle version finally adopted. This has been expanded to include, as mentioned, the use of manual locks, as well as checking procedures to avoid identified pitfalls. The thirty-six-step single-cycle procedure is detailed with illustrations in the **User Manual (2013)**, **Setting up a Calculation**, pp 20-31. The procedure includes the provision for setting up an automatic cycle counter that increments by 1 each time a calculating cycle is completed.

Various checking and diagnostic procedures were developed during the build to systematically verify correct operation and to diagnose suspected faults. Stepwise procedures are described and illustrated in the **User Manual (2013), Section 7**, **Trouble Shooting**, **Diagnostic Tests**, pp. 117-124. These include separate procedures for testing the following operations:

1. Carry warning
2. Carry lever locking
3. Carry lever reset
4. Carriage of tens

Systems checks are described as stepwise procedures for:

1. Addition and Carry System Check
2. Secondary-carry Propagation Test.

10.1 Additional Information

Engine Logs

1. Reg Crick's Logs, Science Museum, London

Throughout the specification and construction of the London Engine detailed logbooks were kept by Reg Crick, the engineer largely responsible for producing the manufacturing drawings and for the physical construction of the Engine. The logs are in desk diaries and notebooks, and chronicle daily activity, problems, solutions, and progress. They contain detailed technical information, engineering insights, and the unfolding rationale for actions taken during the construction. Notable amongst these is 'The Red Book' which spans July 1989 to March 1996. As a contemporary chronicle they are of special interest as events were documented as they happened i.e. in ignorance of what was to come.

2. Technical Log for the USA Engine

The logs contain detailed records of technical issues relating to fault-finding, diagnosis, repair, parts replacement, maintenance and operation. The log includes remedial action taken to repair the Engine after damage in transit to the USA. From May 2008 till January 2016 the Engine was operated, maintained and repaired by a local volunteer team led by Tim Robinson at the Computer History Museum, Mountain View, California where it was on loan for public display and demonstration. The log includes technical issues resulting from repeated operation over a sustained period of time. The logs are deposited in the Computer History Museum archive in Institutional Records, Lot X3711.2007.

Published Works

The following are published works that provide overview, context, and relevant technical and narrative material.

Bromley, Allan G. "The Evolution of Babbage's Calculating Engines." *Annals of the History of Computing* 9.2 (1987): 113-36.

Technical overview of Babbage's Difference and Analytical Engines. Contains a timeline of Babbage's work on calculating Engines.

---. *The Babbage Papers in the Science Museum: A Cross-Referenced List*. London: Science Museum, 1991.

A printed cross-referenced catalogue of Babbage's technical archive. This listing provided the basis for the subsequent re-cataloguing by the Science Museum of the Babbage papers.

---. "Difference and Analytical Engines." *Computing before Computers*. Ed. Aspray, William. Ames: Iowa State University Press, 1990.

Swade, Doron. *The Cogwheel Brain: Charles Babbage and the Quest to Build the First Computer*. London: Little, Brown, 2000. US edition: *The Difference Engine: Charles Babbage and the Quest to Build the First Computer*. New York: Viking, 2001.

Accessible historical account of Charles Babbage's efforts to build calculating engines in the 19th century followed by the modern sequel – a narrative account of the construction of Difference Engine No. 2 at the Science Museum, London, starting in 1985. Ends with the successful completion of the calculating section in 1991. Does not include an account of the later construction of the output apparatus.

---. "The 'Unerring Certainty of Mechanical Agency': Machines and Table Making in the Nineteenth Century." *The History of Mathematical Tables: From Sumer to Spreadsheets*. Eds. Campbell-Kelly, Martin, et al. Oxford: Oxford University Press, 2003. 143-74.

---. "Calculation and Tabulation in the 19th Century: George Biddell Airy Versus Charles Babbage." PhD. University College London, 2003.

---. "The Construction of Charles Babbage's Difference Engine No. 2." *IEEE Annals of the History of Computing* 27.3 (2005): 70-88.

---. "'Photographing the Footsteps of Time': Space and Time in Charles Babbage's Calculating Engines." *Space, Time, and the Limits of Human Understanding*. Ed. Shyam Wuppuluri, Giancarlo Ghirardi: Springer, 2017. 417-27.

---. "George Biddell Airy, Greenwich and the Utility of Calculating Engines." *Mathematics at the Meridian: The History of Mathematics at Greenwich* Ed. Tony Mann, Raymond Flood, Mary Croarken: Chapman & Hall/CRC, 2020. 63-81.

Nineteenth Century

Babbage, Charles. "On a Method of Expressing by Signs the Action of Machinery." *Phil. Trans. R. Soc.* 116 (1826): 250-65. Reprinted in: *Babbage's Calculating Engines: A Collection of Papers: A Collection of Papers Relating to Them; Their History, and Construction*. Ed. Babbage, Henry Prevost. London: Spon, 1889. 236-41. Reprinted in: *The Works of Charles Babbage*. Ed. Campbell-Kelly, Martin. Vol. 3. London: William Pickering, 1989. 209-23.

---. *Laws of Mechanical Notation, Chapter I: On Lettering Drawings*: [Privately printed], 1851. Reprinted in: *Babbage's Calculating Engines: A Collection of Papers Relating to Them; Their History, and Construction*. Ed. Babbage, Henry Prevost. London: E. and F. N. Spon, 1889. 242-57. Reprinted in: *The Works of Charles Babbage*. Ed. Campbell-Kelly, Martin. Vol. 3. London: Pickering, 1989. 224-8.

Babbage, Henry Prevost. "On Mechanical Notation, as Exemplified in the Swedish Calculating Machine of Messrs. Scheutz." British Association. September (1855). Reprinted in: *Babbage's Calculating Engines: A Collection of Papers Relating to Them; Their History, and Construction*. Ed. Babbage, Henry Prevost. London: Spon, 1889. 246-7.

---. "Scheutz's Difference Engine, and Babbage's Mechanical Notation." *Minutes of Proceedings of Civil Engineers*. May (1856). Reprinted in: *Babbage's Calculating Engines: A Collection of Papers: A Collection of Papers Relating to Them; Their History, and Construction*. Ed. Babbage, Henry Prevost. London: Spon, 1889. 248-257.

Videos and Simulations

Five excellent explanatory simulations by Mike Hilton of Difference Engine No. 2 mechanisms https://www.youtube.com/playlist?list=PLSOxgHhh6-o8ZuhRpL9ds8wM4doxauru-

Video of model, by Piers Plummer, of section of Difference Engine No. 2 calculating mechanism driven by a steam engine. https://www.youtube.com/watch?v=t8aYkow-Fv8

10.2 Archive Drawings

The twenty main drawings for Difference Engine No. 2 are part of the series of drawings for the Analytical Engine identified by Babbage with the prefix [A].

The Babbage papers were recatalogued by the Science Museum following digitisation in 2011. The drawing numbers in the list that follows are those used in the new catalogue. The drawing titles are the full form taken directly from the original drawings held by the Science Museum.

In addition to the twenty main drawings listed of which images are provided here, the archive contains drawings of superseded designs, selected derivative tracings, and Notations. These are of historical and technical interest but are not included here as they played no appreciable part in the interpretation of the main designs though there are occasional references to them in the course of the account and the images are available online.

The full archive of Babbage papers is available online at:
http://collection.sciencemuseum.org.uk/documents/aa110000003/the-babbage-papers

Citation convention

The digitized images of the main drawings for Difference Engine No, 2 are identified by descriptors of the following general form:

BAB/X/YYY/Z

where X is a letter (identifying a class or set or related drawings), YYY a three-digit number and Z a single-digit number.

Because of the frequency with which the twenty main drawings are cited in the text the BAB/ prefix has been omitted but only for the twenty main drawings listed here and for which images are included in this Appendix. So in the text BAB/A/163 is cited as A/163. For other than the twenty main drawings the full descriptor is used i.e. including 'BAB/' prefix – this to assist with identifying images online.

The second major class of drawings is that of the Construction Drawings, a set of 219 drawings specifying the 8,000 parts of the machine. All citations of the Construction Drawings are in full i.e. they are all prefixed with 337/ – this to disambiguate them from Babbage's original drawings some of which share the same class number as the Construction Drawings.

Main Mechanical Drawings for Difference Engine No. 2

BAB/A/147 Stereotype frames for Analytical and Difference Engines, January 1847.
https://collection.sciencemuseumgroup.org.uk/documents/aa110000290

BAB/A/159 General Plan and Details of Cams for Driving Calculating Axes also the Disconnecting Apparatus [3]***K*** First Mover [undated].
https://collection.sciencemuseumgroup.org.uk/documents/aa110000302

BAB/A/160 [Elevation] [Untitled, undated. Table of Symbols on reverse]
https://collection.sciencemuseumgroup.org.uk/documents/aa110000303

BAB/A/161 General Plan and Detail of the Driving of the Calculating part of Diff[eren]ce Engine No. 2 [undated].
https://collection.sciencemuseumgroup.org.uk/documents/aa110000304

BAB/A/162 Part of Frame for Supporting Stereotype axes [undated].
https://collection.sciencemuseumgroup.org.uk/documents/aa110000305

BAB/A/163 Elevation of Difference Engine No. 2 (1/4 size) for Plan see No. 2.
https://collection.sciencemuseumgroup.org.uk/documents/aa110000306

BAB/A/164 Plan of Difference Engine No. 2 (1/4 size) [undated].
https://collection.sciencemuseumgroup.org.uk/documents/aa110000307

BAB/A/165 [Figs] 6 & 7 End View and Elevation of Paper Rollers [undated]
[Figs] 4 & 5 End View and Plan of Carrying axis and Driver
[Figs] 1 & 2 End View and Elevation of Supports to end of same Parts in axis [4]$\mathscr{C}$.
https://collection.sciencemuseumgroup.org.uk/documents/aa110000308

BAB/A/166 Apparatus for moving Stereotype frames for Analytical & Difference Engines. Half real size [undated].
https://collection.sciencemuseumgroup.org.uk/documents/aa110000309

BAB/A/167 Elevation and End View of Part of General Framing [undated].
https://collection.sciencemuseumgroup.org.uk/documents/aa110000310

BAB/A/168 Plan and End View of Cams etc for Vertical Motion to Calculating axes [undated].
https://collection.sciencemuseumgroup.org.uk/documents/aa110000311

BAB/A/169 Plan of Cams for Locking Odd Difference Fig[u]re Wheels and for Vertical motion of even Difference warn[in]g [undated].
https://collection.sciencemuseumgroup.org.uk/documents/aa110000312

BAB/A/170 Plan of Cams for Circ[ula]r Mot[io]n of Even Diff[eren]ce Axes [undated].
https://collection.sciencemuseumgroup.org.uk/documents/aa110000313

BAB/A/171 Difference Engine No. 2 Addition Carriage and mode of Driving the Axes [undated].
https://collection.sciencemuseumgroup.org.uk/documents/aa110000314

BAB/A/172 End view of Inking Printing Paper and Stereotype Apparatus [undated].
https://collection.sciencemuseumgroup.org.uk/documents/aa110000315

BAB/A/173 Plan of Inking Printing and Stereotype Apparatus [undated].
https://collection.sciencemuseumgroup.org.uk/documents/aa110000316

BAB/A/174 Rack Pinions for connecting Table fig[ure] wheels with Printing Stereotype Sectors [undated].
https://collection.sciencemuseumgroup.org.uk/documents/aa110000317

BAB/A/175 Plan of Cams for Punching with small stereotype sectors, and Cams for removing Paper Rollers [undated].
https://collection.sciencemuseumgroup.org.uk/documents/aa110000318

BAB/A/176 Plan of Calculating part of Difference Engine with the means of conveying numbers to Stereotype sectors [undated].
https://collection.sciencemuseumgroup.org.uk/documents/aa110000319

BAB [A] 177 Difference Engine No. 2. Bars and Levers for lifting Axes, Plan [undated].
https://collection.sciencemuseumgroup.org.uk/documents/aa110000320

Timing Diagram

BAB/F/385/1 Notation of Units for Difference Engine No. 2, March 1848.
https://collection.sciencemuseumgroup.org.uk/documents/aa110001315

10.3 Construction Drawings

The construction drawings consist of piece-part drawings that specify the manufacturing details for the 8,000 parts of the Engine. The set includes General Assembly drawings showing how collections of parts are combined into mechanisms.

There are 219 construction drawings arranged in thirteen sets:

A	Base and Transport Bracing; and Calculation (Trial Piece)
B	Circular Motion: Drive and Control
C	Calculation (see also A above, Trial Piece)
D	Drive
E	Vertical Motions
F	Framework
G	Mobile Showcase
J	Electronic Printout Mounting
K	Printing
L	Stereotype Printing
M	Modifications to Output Apparatus
X	Trial Piece Control, Timing Diagram, , Matrix Pan Formats, Travelling Platform Control, Mirrored Layout

Miscellaneous: Developmental Drawings

Citation Convention

Part numbers specified in the construction drawings take the general form:

337 P XXZ A

where

- 337 is the project number and is the same for all the drawings
- P is the Series letter of a class or set of related drawings (one of the letters listed above).
- XX is a two-digit number of a sheet that typically specifying several related parts
- Z is a single-digit number identifying a specific part on a sheet (there are never more than 10 details on a sheet).
- A is a detail and/or variation, typically A or B

For example: 337 X 21 is the Timing Diagram in the set of Series X drawings. It is a single view with no additional detail or variation (Timing Diagram is shown on p. 221).

Notes

1. Parts for Calculation are split between Series A and C. The reasons are historical: the Trial Piece was built before the main Engine to verify the altered layout for mirroring of alternate axes. So the Trial Piece required specification of key components for addition and carriage including figure wheels, sector wheels, carry arms and carry levers, and locks. These were the first parts for which manufacturing drawings were produced and the drawings were identified as Series A. The A series-letter was later appropriated for Base and Transport Bracing. The A-series drawings relating to Calculation were not redrawn for inclusion in Series-C, nor were they relabelled. So the content of Series A is anomalously mixed but the Trial Piece drawings are in a their own folder in Series-A.

2. Series J and Trial Piece Control (in Series X) are experimental and speculative without any direct bearing on the machine as built. Series J relates to the mounts for an electronic readout of the figure wheels. The idea was to instrument automatic readout from the figure wheel axis to a computer for checking results. In the event the mechanical computations were found to be more reliable than the electronic verification. The apparatus was implemented experimentally and subsequently abandoned. The Trial Piece Control drawings (in Series X) are plans for automatic cycling of the Trial Piece using an electronmechanical drive intended for unattended visitor operation. This was not implemented. Manual operation for demonstration was preferred.

3. Project number prefix: Rhoden Partners Ltd, a specialist engineering company, was commissioned to prepare drawings for the Trial Piece, a manually operated model to test the layout modification featuring the mirrored layout of alternate calculating axes (Fig. 3.14). Rhoden allocated 337 as the project number as part of its internal numbering system. When Rhoden went into receivership in 1990, the Science Museum hired two Rhoden engineers, Reg Crick and Barrie Holloway, to complete the project in house. The 337 project-number was retained as the project prefix in all subsequent drawings notwithstanding Rhoden's demise.

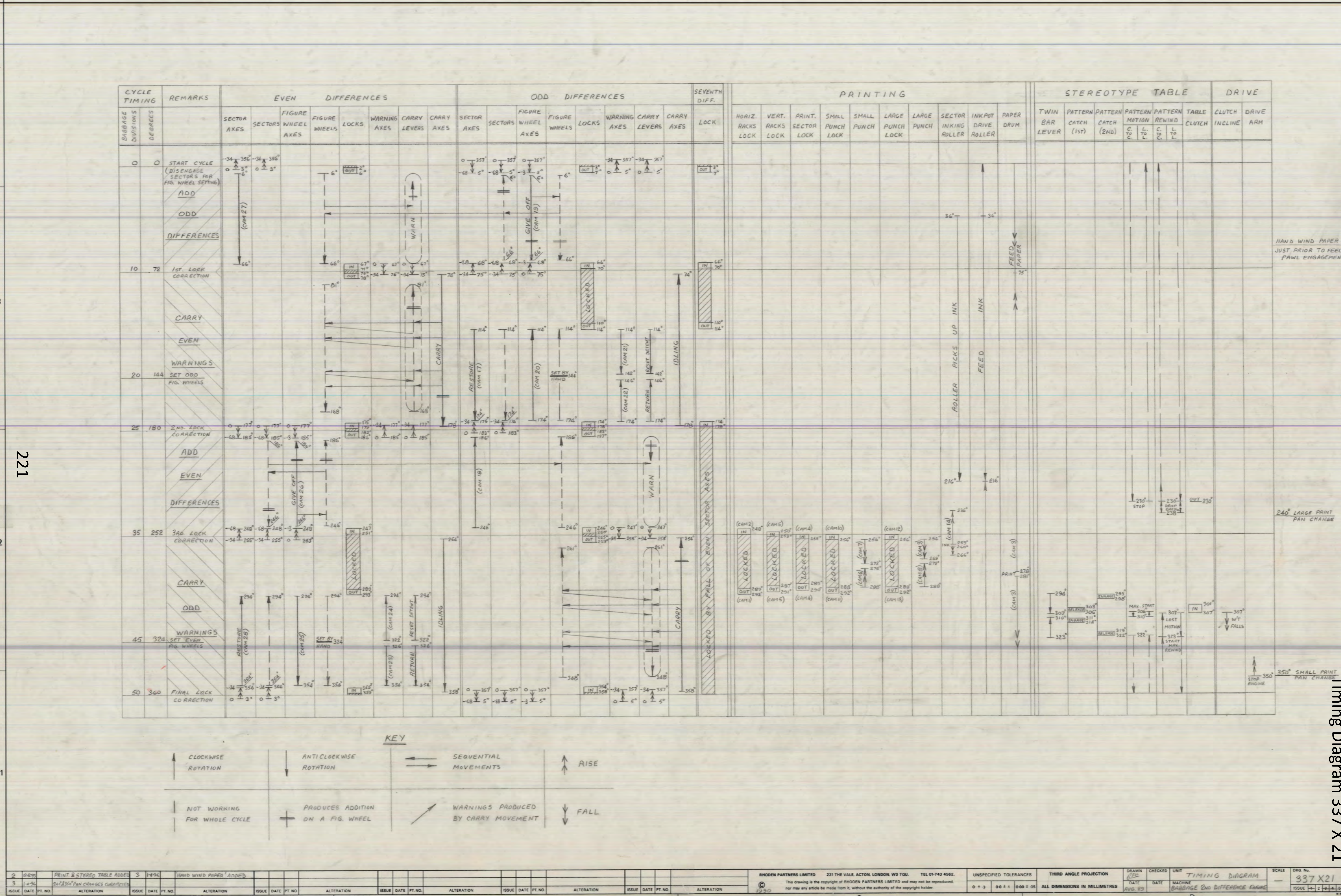
10.4 Parts Lists

Parts lists are used in conjunction with the Construction Drawings listed in Appendix 10.3.

The Parts Lists specify:

1. The part number as identified in the Series drawings.
2. How many of each part are required.
3. Description or name of the part.

There are five Parts Lists for the construction drawings.
The five lists collectively run to 74 sheets of A4.
The Parts Lists were checked and amended in October 2008 after the completion of the US Engine.



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Timing Diagram 337 X 21

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Fig. 8.1 page 198 by kind permission of Tim Robinson.

Frontispiece (elevation of Difference Engine No. 2), Fig. 1.2 page 7, Fig. 4.2 page 52, Fig. 4.5 page 55, Fig. 5.1 page 71 Doron Swade.