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“Hydraulic Packing-Presses.”

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THE packing of cotton piece-goods for export is a characteristic trade in Manchester. The appliances used in the process are of considerable magnitude, though not comparable to the enormous forging-presses now used, and are of interest as an illustration of that gradual mechanical development which is a feature of this century. Probably, however, they have not come under the notice of many members of the Institution.

The number of packing-presses in Manchester is not far short of five hundred, and the accessories to these presses include many separate installations of steam- and gas-power pumping-plant with accumulators and hoists.

In the earliest days of hydraulic packing the press-rams were from 6 to 10 inches in diameter, supplied by hand-pumps, to which power was first applied some fifty or sixty years ago by coupling a revolving crank and the hand-lever with a connecting-rod. Twenty-five years ago the prevailing sizes of press were from 9 inches to 14 inches in diameter, worked at a pressure of $2\frac{1}{2}$ tons to 3 tons per square inch, giving a maximum squeeze on the bale of 460 tons. The only material then available at reasonable cost for making the cylinders was cast-iron, and the pressures used were as great as the nature of the material permitted. The 14-inch press cylinder, with a thickness of 10 inches, was stressed in its innermost layer to over 4 tons per square inch. Frequent breakages demonstrated that the stress was too great. The deficient strength of the cylinder could not be increased by adding more metal, as it was found in practice that if a greater thickness were employed the metal was less homogeneous, and that the increase of strength expected on theoretical consideration was not obtained. The stress could not be reduced by increasing the diameter of the cylinder used for a given squeeze, from considerations of external size which was occasionally limited by the openings into the warehouse basements containing the presses. Whilst this difficulty of providing the pressing power was felt, there was a constant desire on the part of exporters to pack tighter and tighter, so as to ship

more goods in the same bulk, and thus obtain a practical reduction of freight charges. The conditions fixing the limited squeeze were altered by the introduction of steel. This material, after a period of hesitation and doubt, is now thoroughly established for packing-press cylinders, and its use has enabled the maximum gross squeeze to be increased to 700 tons with a substantial factor of safety, as against the old squeeze of 460 tons with a vanishing factor. The present maximum squeeze is not likely to be further increased; already the freight measurement is exceeded by the dead weight, that is, 40 cubic feet measure of bales weighs more than 1 ton, and by some shippers the pressure is considered detrimental to the appearance and quality of the cloth.

The press cylinder is made of cast-steel with an ultimate tensile-strength of 25 to 35 tons; the usual thicknesses are, for a 14-inch cylinder $2\frac{3}{4}$ inches, and for an 18-inch cylinder $3\frac{1}{4}$ inches, giving (by formula $p \frac{R^2 + r^2}{R^2 - r^2} = P$, when P = the tension per square inch on the inner layer, R and r the outer and inner radii, and p the pressure in the cylinder) a maximum stress of 9 tons on the inner layer of a 14-inch cylinder when worked to $2\frac{3}{4}$ tons, and a slightly higher stress with the 18-inch press worked to the same pressure; but, as the 18-inch presses are seldom worked beyond $2\frac{1}{2}$ tons on the square inch, the margin of strength is proportionately increased (*Fig. 1*). Some makers determine the proportions of the cylinders by the average stress, taking the total bursting-pressure on the area of steel sustaining the stress; say, on an 18-inch press, $19 \times 2\frac{3}{4} = 52$ tons on an area of $6\frac{1}{2}$ square inches = 8 tons on the inch—a considerable discrepancy as compared with the 9.6 tons of the formula. The water is admitted centrally through the bottom. The old method of entering through the side of the cylinder led to fractures, and is now discarded. The upper part of the cylinder is bored for a length of 12 to 18 inches to fit the ram, and form its guide, the lower part of the cylinder being cast large enough to clear the ram and give an annular space for the water. A recess, about 2 inches long, varying from $\frac{1}{2}$ inch to $1\frac{1}{4}$ inch in radial depth, is bored in the guide part to receive the leather packing, which is of $\cap$ section. The practice of different makers varies as to the form of the $\cap$; some engineers making the sides of the $\cap$ to touch, others leaving them as much as $\frac{3}{4}$ inch apart. The close $\cap$ seems to give the best results, being stiffer in the bend, and therefore better able to resist deformation and its consequences, namely, increased friction and liability to seize. Proposals have been made to support the bend

with a rigid support, but the close-sided $\cap$ is a simpler method of arriving at the same result. The upper part of the cylinder is enlarged in the usual way to form a supporting collar, which rests on the press bottom.

The pillars, four in number, which act as supports for the press top and as bolts to resist the pressing force of the ram, are forged of wrought-iron, or occasionally steel, turned with great care to an

Fig. 1.

equal length, so that the pressure is taken equally amongst them. The variations of load on the different pillars, due to irregular placing of the goods before pressing, are small as compared with the strength of the pillars, and are neglected in practice. The prevailing diameter for an 18-inch press is 7 inches, giving a stress of about 4.5 tons per square inch. The pillars are usually turned

with swells to receive pressure from the press top and bottom. Nuts are rarely used by Manchester press makers, the pillars being inserted in recesses formed to receive them.

The press tops and bottoms, which are in effect beams, are always made of cast-iron. The tops were originally constructed with a continuous bottom flange or plate forming a plane surface for pressing against, and strengthened by ribs running lengthwise of the top. The beam was thus unnecessarily strong in resisting compression, but weak as against tensile-stresses; many failures resulted from this arrangement of metal, and the form was altered, first, the Author believes, by Mr. John Hopkinson, by the introduction of a continuous top flange, thicker than the under flange. A compound beam is thus formed, of which the proportions have been determined by practice. If a maker found his press tops to fail he strengthened them in the places indicated by the fracture, and thus gradually evolved a form which survives. The press bottoms had a somewhat similar history; the bottom continuous flange was early introduced for the press to rest on, and the difficulties were rather with the replacement of the strength which was taken away by the large central hole for receiving the cylinder. With cast-iron cylinders this was required to be from 30 inches to 36 inches in a 14-inch press. The pillars are held in the press top and bottom in bored sockets, with openings and cover-plates; faced projections enable the pillar swells to bear normally and equally.

The ram acts under simple compression, and is made very massive. To avoid corrosion, and to reduce friction, the upper part is usually cased with a turned and bored brass hoop from 42 to 48 inches long, or covered with copper by electric deposition, a method introduced by the Broughton Copper Company under Wilde's patents. The table or platen has had a history somewhat similar to the press top. Failures have led to increased proportions, and now in the larger presses it is of very substantial dimensions. The table is guided by four rollers, for which the turned pillars form a path. Rollers are fixed under the table to facilitate the removal of the table for renewing the packing leather.

The various methods in use for supplying the water under pressure to the presses, may be divided into two classes:—

1. Engines running at constant speed, and driving independent pumps.
2. Direct-acting pumps.

The majority of the older plants have a pair of high-pressure non-condensing engines, running nominally at constant speed,

cutting off at from one-third to two-thirds of the stroke, and driving by spur-gear from the crank-shaft on to a second-motion shaft, which drives the pumps through spur-gear (iron and wood). The pumps, usually 2 inches in diameter by 4 inches stroke, are arranged in sets of six, worked from a crank-shaft driven by gearing, and with the cranks set at intervals of 60° .

The delivery-valves are usually weighted to deliver at low pressure from all six pump-barrels, and to knock off successive pairs as the pressure rises. In large plants, where several sets of pumps are necessary, the high-pressure water is pumped by one set of pumps, the others knocking off altogether when their limit of pressure is reached. This system of pumping is also used in combination with accumulators, in which case the majority of the pumps knock off when the accumulator is full, the finishing squeeze being given as before by the pumps direct. The cost of repairs under this system, and the loss of power in starting and stopping the fly-wheel, and transmitting the motion to the pumps, have led to the general adoption in the newer plants of direct-acting pumps, in spite of the apparent advantage in the older system of using the steam expansively.

The direct-acting system was first applied by Messrs. Nasmyth, Wilson and Co., at the warehouse of the late Mr. Mendel, and this plant presents features peculiar to the installations by that firm; great pumping-power is required without a reservoir of pumped water or fly-wheel momentum. The steam is used without expansion, and the pumps always run when any valve by which pressure water is drawn from the pipes is open. The speed is regulated by the resistance, and is limited by throttling the steam-supply with the stop-valve. When no valve is open the engine stops, or creeps round very slowly if there be any leakage. The steam is always on in the cylinders, and of course there is perceptible loss by condensation. The pumps in this particular case are 2 feet $1\frac{3}{4}$ inch in diameter by 22 inches stroke, with steam cylinders 20 inches by 22 inches, set at quarter-centres, but in later plants the pumping-power has been largely increased.

Other forms of direct-acting pumps without fly-wheels have also been used without accumulators, such as those of Messrs. Tangye and Worthington; but a more common form is to keep so far to the old lines as to have a fly-wheel for turning the centres, and not to trust to the valves to reverse the pumps. The accumulator system, first adopted by Messrs. John Shaw and Sons in connection with packing-plants, very much diminishes the pumping-power required for a given speed of pressing a bale, and averages the load to some

extent; further, by using the accumulator as an engine-regulator, the losses of running at full speed, when little or no work is being done, are very largely diminished.

The standard accumulator pressure is 1,120 lbs. to the square inch, and with this pressure sufficient force is provided for raising the press-table through from 78 to 88 per cent. of its range, compressing the bale to a considerable extent. The final squeeze is given by the pumps. There is occasionally a waste of accumulator water if a plant is not fully occupied. Suppose that the accumulator is filled by the pumps before the bale is completely pressed or "set," water must be let off from the accumulator to keep the pumps going for the supply of high-pressure water.

The convenience and saving of time in working a plant with an accumulator are undoubted, but the arrangement as ordinarily worked is not economical, as water which has been pumped to $\frac{1}{2}$ -ton pressure is used for lifting the ram, table and goods, for which 80 or 100 lbs. would be ample. To prevent some of this waste, and to secure the advantage of pumping against a constant pressure, Mr. Bellhouse has introduced an intensifier and diminisher,¹ which, in its original form, was similar to that described by Mr. R. H. Tweddell,² but is now adapted to work first as a diminisher, producing water at 224 lbs. pressure for the early stages of the lift of the ram; and secondly, after the accumulator $\frac{1}{2}$ -ton pressure has continued the pressing, another stage coming in as an intensifier to give $2\frac{1}{2}$ tons pressure on the square inch.

The relative economies of the various ways of pumping are best compared by examining a diagram of the pressures corresponding to the travel of the ram whilst pressing a bale. In *Fig. 2*, curves A^1A^1 , A^2A^2 , represent the movements and pressures whilst packing a bale of seventy pieces, of which sixty were first pressed, the press then being lowered and ten pieces more added; BB represent similarly the compression of a bale of sixty-five pieces; and in *Fig. 3* CC represent two successive bales of another class of goods. The following Table results from the diagrams, neglecting fractions of a foot-ton, and the large reverse curvature from 28 to 31 inches in one of the CC curves, which is evidently due to the 10 cwt. accumulator-pressure coming on too quickly:—

¹ See Minutes of Proceedings Inst. C.E., vol. xciii. p. 567, for a similar arrangement in crane work.

² Institution of Mechanical Engineers. Proceedings. 1874. p. 168.

Fig. 2.

RISE OF PRESS-TABLE AND CORRESPONDING GAUGE-PRESSURE.
A1 160 pieces, A2 160 pieces and 10 unpressed pieces, B1 65 pieces.
Vertical scale gives pressure, horizontal scale movements of table.

Fig. 3.

RISE OF TABLE AND CORRESPONDING GAUGE-PRESSURE FOR TWO SUCCESSIVE BALES.

	Net Work.	Net Work used from Accumulators and Constant-power Pumps.	Net Work used by Diminisher, Accumulator, and Intensifier.	Gross Pump-power varied to suit Load.	
				80 per cent. efficiency.	75 per cent. efficiency.
	Foot-tons.	Foot-tons.	Foot-tons.	Foot-tons.	Foot-tons.
A ¹ A ¹	225	714	466	281	300
A ² A ²	68	411	186	85	91
A ¹ A ¹ + A ² A ² .	293	1,125	632	366	391
B B	258	730	461	323	344
C C	325	843	618	406	433

The time occupied in a single compression, such as BB or CC, varies with the character of the goods and the power of the plant; about one minute would be the minimum, the corresponding average rate of work used would be 57·2 indicated HP. It will be seen from the Table that the best results are very wasteful, and that a system by which the water could be pumped at the pressure at which it is used would require a smaller net power to be delivered as compared with the best present results from direct-acting pumps, and, if combined with an economic use of the high-pressure steam in the generator, would probably amount to 60 per cent. of saving. The difficulty which prevents this saving is the irregularity of the work. The time of pressing is about one-ninth of the whole time of making a bale, so that a plant of at least nine presses is necessary for constant running if multiplication tended to a true averaging, but it is not so. The tendency is for presses to get into step with each other. Of two presses rising together, the press with the less pressed bale in it offers least resistance, and takes the water until pressures are equalized.

The other remedy suggested to this wasteful working is in a development of the "intensifier and diminisher" principle. If arranged to be adjustable to the pressure, a constant-pressure supply would give the varying pressure required. To produce such a differential intensifier is a problem for the inventor, who will need to remember that the convenience of the wasteful system, and its economy in the time of the packers, will enable it to maintain its position unless very great improvements in mechanical efficiency are arrived at.

The magnitude of the pressures prevents the convenient deter-

mination of the friction at high pressure; but for low pressure, depending entirely on the weight of the ram and table, it is easy to obtain approximate results by gauging the pressure with the ram when rising and when falling. The differences so obtained by the Author vary from 2 to 5 per cent., giving a coefficient varying from 1 to $2\frac{1}{2}$ per cent. Taking into account the circumstances of the various readings, the Author would place the actual coefficient for low pressures and low speeds at about 1.25, or rather higher than Mr. Tweddell, who gives 1 per cent. as the loss by friction in using water-pressure in well-constructed plant in good order. For high speeds, the friction in the pipes and the difference between cylinder-pressure and gauge-pressure prevent any accurate determination in this way.

Steel piping is now always used for the high-pressure supply, not only because of its superior strength, but also for its greater power of withstanding corrosion and furring by oxidation. With accumulator plants, separate piping is used for coupling the accumulator and presses with a simple stop-valve to each press; the higher pump-pressure is regulated by a double-stop or three-way valve, which couples the high-pressure supply to the press or allows the water to pass on to the piping beyond. The same stop, by another handle, couples the press to waste.

The Paper is accompanied by three diagrams, from which the *Figs.* in the text have been prepared.