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" ASSEMBLY OF LOCOMOTIVE PARTS,"

BY

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The object of this paper is to consider the assembly of Locomotive parts, which is performed in the Machine Shops, before they are handed over for erection.

As the Wheels are a very important part, we will consider these first. The assembling must be of a very rigid character, owing to the shocks to which they ere subjected.

Force fits are universally employed for wheels and axles, and shrinkage fits for the tyres. The fundamental principle is the same in both cases, the bore of the outer member is smaller, and the diameter of the shaft or inner member, larger than the finished fit, therefore, the inner member is compressed, and the outer expanded, and the elasticity of the metals produces a radial pressure at the contact surfaces of the fit. The difference in the diameters of the two members is termed the allowance. There are several types of force fits. The principle ones are parallel, taper, and double cylindrical. The foremost is the most reliable for Locomotive wheels, as it is capable of withstanding more shocks. It requires more work to force the axle home than either of the others, and consequently it would require more to remove it.

The taper fit is more difficult to produce, and is liable to become quite loose with a slight movement of the axle in the wheel hub, thereby rendering it unsatisfactory for Locomotive wheels, but it can be employed successfully where the fit has means other than the friction between the contact surfaces to prevent movement, such as the piston rod and crosshead.

The double cylindrical fit is reliable, but like the taper fit it has a disadvantage in regard to production, but it is very useful where a long fit is required, as it does not need so much work to be performed in the assembling, as the parallel fit.

The parallel fit is used at Swindon for axles and crank pins. The wheel hub is bored parallel, smaller than the axle, by the allowance which varies from .0015in. to .002in. per inch of diameter. The wheel is forced on the axle by means of a hydraulic press constructed for that purpose. The tonnage required varies from 8 to 12 tons per inch of diameter, and half the total tonnage should be obtained when the wheel is half way on. An instrument is employed to record the work done throughout the operation. The axle number is then marked on the chart, which is kept for future reference.

With regard to force fits, there are a few practical points which need consideration. If the allowance should be excessive, the elastic limit may be reached and permanent set occur, and, in extreme cases, the ultimate strength of the material passed and the hub burst. In some cases where the hub of the wheel is large in proportion to its bore and the allowance excessive, the metal will flow in front of the axle instead of the hub expanding the full amount thus reducing the effective allowance. The tonnage is usually low in these cases, owing to the point of the axle preventing the other portion from coming into contact with the hub. The following are actual examples : three wheel centres were forced on axles with 9in wheel seats. No. 1 had an allowance of .039in., No 2 .026in., and No. 3 .017in. The tonnage taken to force the wheels home was respectively 65, 50 and 100 tons. Nos. 1 and 2 were taken off and forced on again in the same positions. The tonnage taken the second, time was 105 and 85 tons respectively, which proves that the allowance of .017in. was practically correct and the elastic limit was not reached in either case, but the effective allowance on No. 1 and No. 2 must have been reduced.

The same allowance is not suitable for all makes of Wheel Centres, owing to the difference in the strength of the material, which is found out by trial.

When a force fit fails, in that the tonnage is too low, it is always best to find out the cause. In no case should the allowance be increased until it is proved to be necessary. There are several factors which would be likely to cause failure, such as a taper bore in the hub, the axle large on the point, or excessive allowance. One important point to remember is that the axle should coincide exactly with the bore of the hub, whether it is a parallel or taper fit, because it is the friction at the contact surfaces which produces most of the resistance, not the expansion of the hub.

The design of the wheel is important as the intensity of the grip depends a great deal on the thickness of the hub. The best results are obtained when it is not less than the radius of the axle throughout its entire width.

In forcing the axle into the wheel hub the metal is so disturbed that it requires several days to settle down. If the wheel is allowed to remain on the axle for two or three days, in some cases it will take as much as 100 per cent. more tonnage to remove than it did to force on.

Lubrication of the contact surfaces is necessary or abrasion would occur. A lubricant of a heavy nature should be used, such as white lead and boiled linseed oil, or graphite grease. The latter has proved to be very satisfactory and it has an advantage on account of its being non-poisonous. After the wheels are forced on the axle, the keys are fitted. Care should be taken to ensure a bearing on the sides as well as the top and bottom, to prevent any turning movement which would throw the crank out of quarter. The taper of the keys is 3-32in. per foot. They are fitted to within 6in., and then driven home with a tup.

The wheel centres are then turned to take the tyres which are shrunk on. The allowance for shrinkage is $\frac{\frac{17}{16}D + 10}{1000}$

The tyres are heated in a circular Gas Furnace. The wheel centre is then placed in position and the retaining ring fitted. The lip of the tyre is then knocked down by a power hammer constructed for that purpose, after which it is allowed to cool and, when cold, the tyres are turned. The wheels and axle should be assembled before the tyres are shrunk on as the shrinking on of the tyres reinforces the wheel hub to a considerable extent. When the tyre is on, it requires about 25 per cent. more tonnage to force the wheel on the axle.

The next operation is to bore the crank pin holes and force in the crank pins. The journals of the crank pins are finished in a quartering machine to ensure their correct relationship with one another.

Before leaving the subject of wheels we will consider the assembling of the built-up crank axle.

The built-up crank axle has superseded the solid type at Swindon wherever it is possible to fit one.

In assembling, shrinkage fits are employed, as it is the only satisfactory method to ensure rigidity where the length of fit is short in comparison with the diameter of shaft. The practice at Swindon is to plane the webs to within 1-16in. of the finished dimensions and bore the holes parallel, smaller than the shaft by the allowance.

The shafting is rough turned and a hole bored up the centre after which it is heat treated. The shafts are then turned to the finished dimensions at the fits only, the other portions are left large to be finished after the crank is assembled.

In assembling, the webs are heated in a gas furnace and placed in a special fixture which has means to set the throws in their correct positions. After the crank is cooled, two holes are drilled and tapped, half-way into the web and half-way into the shaft, and screwed dowels fitted. The crank is then ready for finishing.

When assembling shrinkage fits, care must be exercised over the cooling process. The web should be cooled on the side which is set correctly with the end of the shaft, so that it will grip the shaft and contract towards that side.

One has often seen mistakes made when shrinking a loose collar on a shaft against a fixed collar, which was thought would help to obtain the desired result, but this is always unsuccessful as the collar grips the largest part of the shaft and contracts away from the fixed collar. (A slight taper is given to the shaft towards the collar.)

A shrinkage fit is much more rigid than a force fit. In Professor Wilmore's tests the average resistance to slip was for an axial pull 3.66 times greater than that of the force fit, and in rotation or tension 3.2 times greater. In each comparative test the dimensions and allowances were the same.

The piston and rod are other parts that need careful assembling. Some Railway Companies experience considerable trouble owing to the piston working loose while in service.

On the introduction of the box piston, the same trouble was experienced at Swindon, but after experimenting with several types of fits, a very satisfactory one was found.

The types may be divided into two classes : One, in which a collar on the rod is forced against the piston by means of screwing the rod into the piston, and the other, in which a taper fit is used, either plain or screwed. The latter is the standard fitting at Swindon.

The piston is bored and screwed six threads per inch to a taper of 1 in 48. The rod is turned and screwed to fit within $\frac{1}{4}$in. of its final position. It is then forced home in a lathe adapted for that purpose.

The piston is held in a chuck, and a clip fixed to the rod ; one end of the clip being allowed to rest on a plunger of 3in. diameter in a cylinder containing oil, and a pressure gauge is connected to the bottom of the cylinder. The lathe is set in motion, which forces the rod into the piston. The resistance is recorded in lbs. per sq. inch by the gauge. The lever acts at a radius of $7\frac{1}{2}$in, The pressure allowed for 18in. and $18\frac{1}{2}$in. piston is from 950 to 1,000 lbs. per sq. inch. After the rod is forced home a $\frac{1}{2}$in. screwed dowel is fitted half-way into the piston and half-way into the rod. The piston is now ready for finishing.

The piston is allowed a clearance in the cylinder of .002in. per inch of diameter.

The success of the screwed taper fit is due to the large area of surface contact, and it brings into use the elasticity of the metals.

The piston rod and crosshead are the next items for consideration. The taper fit has been universally adopted, but in the fitting of the cotter, the practice varies.

It is usual to allow a draw in the cotterway, that is, one edge of the cotter takes a bearing against the piston rod, and the other against the crosshead. This method provides a means of tightening the crosshead if it should become loose while in service.

It was found that with the introduction of increased steam pressure on the Swindon engines, trouble was experienced through the cross-head working loose, and the cotters bending. This was overcome successfully by discarding the draw in the cotterway. The Swindon practice is to bore the hole in the Cross-head to a taper of 1 in 21. The piston rod is then fitted to within $\frac{1}{8}$in. of the bottom of the hole.

After the cotterway is milled out, the rod is forced home by a hydraulic press designed for that purpose. The end of the rod must butt against the bottom of the hole in the cross-head. The cotterway is broached out to a taper of 1 in 30 by a broaching machine which has a fixture for that purpose. When the cotter is fitted it is practically a solid job.

The coupling and connecting rod bushes are other cases where a force fit is necessary. The holes in the rod are bored parallel and the bushes turned larger by the allowance, which is .002in. per inch of diameter. The bushes are forced in by a Lucas power press. The tonnage required for a bush of 7in. diameter is approximately 12 tons.

A steel set-screw of $\frac{3}{4}$in. diameter is fitted through the eye of the rod and into the bush. The end of the screw is left $\frac{1}{8}$in. below the white-metal surface, so that it cannot come into contact with the crank pin.

The running clearance given for bushes is .002in. The best way to fit a collar bush in a rod is to arrange for the collar to be on the wheel side of the rod, as the bush does not contract so much at the collar side. This is a great help when fitting the rod on the crank pin.

DISCUSSION.

In opening the discussion, the CHAIRMAN (Mr F. W. HAWKSWORTH) referred to the force fit and the shrinkage allowance, and said the Author had stated that the two types of fit are identical, but as a matter of fact, if the Newall tolerance is considered and the tolerance plotted against the diameter of the pin or shaft, it will be found that the curve formed is parabolic, that is, the allowance depends upon the square of the diameter of the pin. In shrinkage fits, according to the formula he had given, the shrinkage allowance depends only on the diameter instead of the square. Another point was with regard to the tyre fastening. No doubt one of the disadvantages of studded tyres is that a fracture is likely to start from the bottom of the hole, and the chief objection is that if two fractures occur between two studs, the piece of tyre is likely to leave the wheel. This led to a serious accident some years ago, and it is easily seen that it may be so. With the key ring, however, the tyre is held through the whole of its circumference. Regarding the method of securing the crank pins to the webs of the built up cranks, they are now secured by round keys. He believed the first lot were secured by flat keys, but he thought the flat keys showed a tendency to work out, therefore they were replaced by screwed dowels. With regard to the apparatus for screwing the piston rod into the piston head, he took it the pressure depends partly on the diameter of the piston rod and also on the class of piston head. Obviously a box piston is stronger to resist the screwing in of the taper than a plain piston such as that in use on the 2301 class engines. The pressure must depend not only on the diameter of the piston, but on the form and design of the head itself. Another point : Was the head lubricated before the rod was screwed into the head, and also what lubricant was used?

Replying, the AUTHOR said he quite agreed regarding the key ring being a protection against the tyre coming right off in cases of fracture. With regard to the crank webs, he said two screwed dowels were fitted in case the web should move. With regard to the piston rods, the pressure varies according to the strength of the piston. With small pistons like those on tho standard goods engines, the pressure is much lower than or larger engines. As to the lubricant for the piston rod soluble oil is used. Regarding forced fits and shrinkage, the principle being the same, the Newall standard gave .002in. per inch of diameter. He did not know what standard the Chairman had seen which plotted a parabolic curve. He had a chart which gave .002 per inch of the diameter for force fits. The principle is the same in each, relying upon the elasticity of the metal. The hub is expanded in each case, but by different methods.

Mr. S. R. Jones referred to the question of lubricant used, and said they tried to avoid anything in the form of grease, the idea being to apply the lubricant so that immediately it is heated it will disappear. It is necessary to use a lubricant of some sort, but they had found from experience that any form of Rape oil, which is a vegetable oil, would have a tendency to affect the cast iron. At the present time they used a little soluble oil, which contains very little fatty substances, and this had proved very successful. Of course, he said, the taper piston fastening is very important. Many years ago, they experienced an enormous amount of trouble with loose piston heads. Many ideas were tried. They were fitted up against a collar, and a buttress thread was tried with a special ring at the back. He remembered in 1906 they had considerable trouble through the slacking of piston heads ; he had seen as many as seven or eight in a week come in with loose rods. This lead to a trial being given to the taper thread, and he (the speaker) supervised the fitting of the first one. It was a steel head and a steel rod, and after it had had a good trial the method was finally adopted. Cast-iron box heads came on after the taper fits were established. There were great doubts as to whether only the thread in the cast iron would, be sufficient to hold it, and certain experiments were made. He said the Author was correct when he said that the pressures vary. With some types of pistons the pressure was 7001b. per square inch, and on the Box heads from 950 to 1,000. Since the pressure gauge had been used for putting them on, which was in 1914, he did not think there had been a single rod go loose with the taper thread. Before that there were just a few, and. in every case he could trace it to the fact that the rod had seized before it was properly home. That was one of the dangers, of the taper thread, but if a pressure gauge was used that danger was much decreased. If a seizure does take place the head should be destroyed at once and no risks taken. These are possible with the different kinds of metal with which one has to contend. If the rod is pulled up and the pressure shown on the gauge increases uniformly, it is certain that the job is correct. He did not know of any loose piston heads at all since 1915, and he did not think any other Railway Company could show such a record as that.

Of course, the fastening of a Piston head is an important thing for more reasons than one. A strong fastening may be made, but water in the cylinder and other things, like that have to be contended with, and he considered it was water which often started the heads becoming loose. If it is up against a collar that can give, one may be fairly sure that before long there will be a loose head. The whole thing has to be taken out and it means putting a new rod in with a box head, as no attempt is made to put an old rod into a new head because it is impossible to guarantee to pull up to the exact length, as the density of the steel has to be considered. With the taper fastening it is certain that the head will not get loose.

Mr. J. T. HILLIARD said he hardly agreed with the figure .002in. per inch of diameter tolerance. He thought that was excessive. He thought it should be according to the size of the boss.

The AUTHOR said that the allowance, of course, depends upon the shape of the wheel, and also the thickness of the boss, but the Newall standard was very satisfactory. It gives .002in. per inch. He had had a long trial of wheels generally and found it very satisfactory on wheels with a 9in. bore, and it was possible to get 100 to 120 tons pressure.

Mr. A. T. CHEESLEY, referring to the previous question, said the Author had remarked that it is proportional to the diameter. According to Newall, he believed the terms given were " Proportional to the square root of the diameter." He did not think it proportional to the diameter. With reference to the allowance being too large and the effect of the large allowance on the hub of the wheel when being pressed on, is it the case, as stated in the paper, that the stress in the metal of the wheel exceeds the elastic limit? If the stress in the metal exceeds the elastic limit, is the permanent set a measurable quantity? He had tried to measure this, but had not been able to do so. No flow of metal of the wheel-boss had been discernible, and at the moment he could give no other explanation but that the elastic limit of the metal had been exceeded. Regarding force fits and the question of lubrication in these, there was obviously a slight difference between a forced fit on a piston rod and an axle, but in one case we are told that we must have no form of lubrication. In the case of the axle it was very essential. Could the Author enlarge somewhat on the question of lubrication of forced fits. He did not think it was quite fair to abandon entirely the buttress thread in the piston rod. He would like to ask the Author which was the best for machine shop practice, to have the allowance on the axle or male part only, or to have it on both, particularly where one is dealing with mass production.

The AUTHOR, replying to Mr. Cheesley, said that with regard to the allowance, he had seen very elaborate formula with reference to the forced fits, and he had seen them only half-a-thousand per inch of diameter. It depended a great deal on what was required. Speaking of a wheel hub of 8in. or 9in. diameter, from experience he thought .002 per inch of diameter very satisfactory, and Newall's standard is approximately .002 per inch of diameter, It depended, of course, on the elasticity of the metal, but he did not see why it should not increase with the diameter, seeing that the elongation of the metal is proportioned to the length approximately. With regard to the permanent set, he thought it quite obvious that if a pin or axle is pressed into a wheel and if the axle or pin be taken out and pressed in again, and should the tonnage increase, he did not think that a permanent set had taken place, otherwise the tonnage would be reduced. With regard to lubricants for forced fits, he said there must be a lubricant of some description. Referring to the lubrication of pistons, he would ask Mr. Jones to answer that. Speaking of the buttress thread it has no advantage over the V thread, for the piston and rod, and it is more difficult to manufacture. If the two tapers should differ the piston would work loose. He thought the V thread was the best.

Speaking of lubrication, Mr. JONES said Mr. Cheesley must have misunderstood him. He did not say that lubricant was not required. What he said was that they attempted to find a lubricant that would disappear immediately heat was applied to it. What they used was one with very little fatty substance in it. His experience was that with cast iron, any oils that contained vegetable matter, are more or less dangerous. When a start was first made sal-ammoniac was actually used to make a tight job, but that was not altogether a great success. He did not say that lubricant was not required, but that you want something you can get rid of when the rod is properly home.

Mr. K. J. COOK said he should like to ask if with the stud fastening in tyres which other Companies use, they scrap their tyres before they are worn down so far as ours, owing to the studs. Also with regard to the clearance in forced fits, did he understand that the allowance varies as the square root of the diameter, that the number of thousandths tolerance is twice the square root of the diameter? With a 9in. shaft it is going to be merely an allowance of .005 as against .018, surely a great difference. .005 must be on the short side.

Replying regarding the scrapping of tyres, the AUTHOR said he did not think other Railway Companies scrapped their tyres any sooner. The thickness for scrapping was universal. With regard to the allowance, he said that in his experience .002 had been very satisfactory. He had never been able to get 120 tons with a .006 allowance. He did not think it was possible.

The CHAIRMAN said that Newall's tolerance was for pins in a fairly solid object, but perhaps it did not apply strictly to such a thing as an axle fitted in a wheel boss, and more or less allowance for axles had to be made and settled in each according to the design of the wheel. This had turned out to be so, by their experience in the G.W.R. Loco. Works. In the early days of the large engines the diameter of the wheel boss on the outside of the wheel was small compared to the inside. It was found that with that shape of boss, instead of the. pressure gradually rising it fell after the axle got about half-way through, owing to the back of the boss opening through there not being sufficient metal. The Bosses are now made almost parallel. He thought the allowance for pressing on, also the final pressure, depended largely on the shape of the boss of the wheel. Therefore, all these pressed fits in the case of wheels have to be treated as special cases.

Mr. HILLIARD said that their experience at the Carriage Works was that with a 7in. seat, 6½in. down to 5½in. diameter, an allowance of .006 was used. With more than that the required pressure was not obtained, the thickness of the boss being 3in.

The AUTHOR said that with wagon wheels 6in, diameter and 7in. length of fit it was possible to get up to 80 tons, but if an attempt was made to get 100 tons trouble would be experienced. With engine wheels 100 tons. was quite easily attained.

Mr. CHEESLEY said there was a lot of difference between the boss of an engine wheel and the boss of a wagon wheel. The boss of a wagon wheel was flexible as against the stiff boss of an engine wheel. He asked the Author's experience regarding the speed of pressing on. Some people had been having trouble through trying to press on too quickly.

The AUTHOR said that if the operatives pressing them on are watched, it will be noticed that they vary the speed according to how the pressure is acting, and when fresh operatives start they generally get failures through being in a hurry. The slower the metal is disturbed, the better the results obtained. If pressed on too fast trouble is experienced, as the wheel is not given a chance to settle down on the axle. It was not possible to lay down any hard and fast rule with regard to pressing on wheels. It all depended on the design of the wheel, etc. It was impossible to get two wheels the same, as the strength of the material was not the same. An extra spoke in the wheel would alter it, while two or three extra spokes would alter it by several tons. He spoke of two pins being pressed in. The first time it took 7.51 tons to press in. It was allowed to remain in for two days, and it took 16 tons to press out. The same pin was pressed back into the hole again and a pressure of 17 tons was recorded. There was no doubt that the best time to press the wheel on was with the tyre off. The tyre reinforces the wheel a great deal. With the tyre on, it takes 25 per cent. more pressure than with the tyre off.

Mr. K. J. COOK asked if, with the axles referred to, the clearance was measured after they had been on once.

The AUTHOR replied, that he had mentioned that the effective allowance was reduced. The stronger the boss the more difficult it is to expand it, and if allowance is excessive the metal will flow in front of the axle.

The Chairman said that he had mentioned just previously that the tolerance allowed for the wheel boss depended largely on the design of the wheel. That seemed to be borne out by the discussion. The question of the shrinking on of the tyres was also a matter for practical experiment, and the formula they now used had been arrived at owing to the fact that there had been several cases of tyres slipping round the wheels when the old shrinkage allowance had been used. The old allowance was .042in. for a 20in. wheel centre, increasing to .078in. for a 90in. centre. The movement of the tyres was thought to be due to the heating up of the brake blocks. They heated to such an extent that the shrinkage allowance was lost and the tyre slipped round. This led to experiments being carried out with their shrinkage allowance, and to investigations with the shrinkage allowance laid down by the American Master Mechanics' Association. This allowance increases rapidly with the diameter. The present allowance, when plotted against the diameter gives a graph something between the old Great Western allowance and the American Master Mechanics' Association. As the diameter increases, the allowance becomes much greater than it used to be, but at the same time, less than the American Master Mechanics' Association. At the present time that shrinkage allowance seems to give every satisfaction.

The Author said that on their 6ft. 8½in. tyres the stress works out somewhere about 36,000 lbs. per square inch, and the total resistance to slip or pull was about 350 tons, which went to show that they were easily within the elastic limit of the tyres.