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The Canadian Engineer

A weekly paper for Canadian civil engineers and contractors

THE PAVING OF STREET RAILWAY TRACK AREAS

ARTICLE DEALS WITH NECESSARY CHANGES IN PAVING TRACK ALLOWANCES CAUSED BY INCREASING WEIGHT OF CARS AND GENERAL TRAFFIC CONDITIONS.

By L. McLAREN HUNTER, C.E.,
City Engineer's Department, Ottawa.

THERE is probably no other public utility which has undergone a greater evolution in the last twenty years than the street railway. During the last few years of development the cars that were used averaged about five tons in weight; and the pavement in the track areas was designed accordingly.

In the past few years the development has been extremely rapid and the weight of the cars has jumped from ten tons to the double-trolley, 25-ton cars, which are to-day used in Ottawa.

The writer intends to give in this article details of the various kinds of pavements constructed in Ottawa on the street railway area, showing the evolution from the time the pavement was designed to stand the five-ton car.

Fig. 1 shows the type of pavement which was first constructed in Ottawa in the track allowance; this was about twenty years ago, and happily only two streets were laid in this manner. Before these streets were repaved the whole track area had deteriorated due to the vibration of the heavy street cars. The two sandstone blocks that were laid in the margin next to the rails had sunk about six inches and the asphalt between the blocks was so badly cracked that passing vehicles striking the rails or blocks with their wheels lifted sheets of the asphalt. Before the debenture period had expired the city had spent enough money on repairs to construct a new pavement.

Fig. 2 shows the next style of pavement tried in the track area. It will be noticed an extra body of concrete was placed under the rail 18 inches in width by 8 inches in depth (from the web of the rail). This type was discarded in 1912 owing to several weaknesses which developed within two years of the pavements being constructed. The principal weakness developed in the two outside blocks next to the asphalt and also for about 12 inches on the asphalt next the blocks. Fig. 3 shows how the pavement deteriorated. It will be observed that the two margin blocks began to sink an appreciable distance below the rail—the asphalt following suit. This was due entirely to the foundation not being strong enough to support the heavy city and interurban cars. The vibration of these cars began to shake and crack the concrete at the end of the wood ties, as shown in Fig. 3, at the point marked "A." The vibration continued until the concrete crumbled away under the blocks, with the result as stated above. Fig. 4 shows a photograph of a dangerous sinkage on this style of pavement.

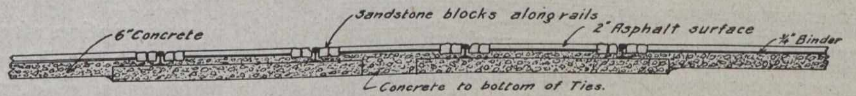


Fig. 1.

Scale: 3/8" = 1'

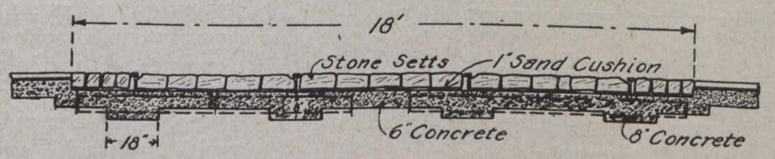


Fig. 2.

Scale 3/8" = 1'

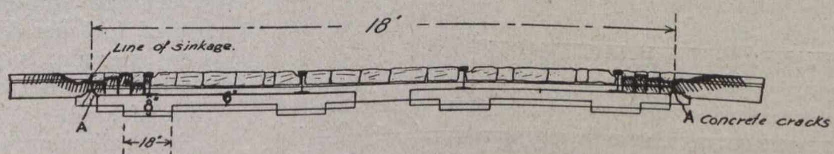


Fig. 3.

Scale: 3/8" = 1'

In 1914 the type of construction for the track allowance was radically changed from former types. Bank Street, one of the main business thoroughfares, was to be repaved and it was done as shown in Fig. 5. An 8-inch concrete slab was first laid 21 feet wide; on the top of this slab was laid a 1-inch cushion of asphalt macadam for the ties to rest upon (this cushion having greatly reduced the noise of the cars). The writer had the railway company bevel the ends of the ties, as shown on the section. This was tried as an experiment to do away with the sinking of the outside blocks—a bad feature of former pavements, as already explained. It will be seen that with the bevelled ties a much heavier body of concrete is between the outside blocks and the ties, thereby preventing any chance of cracking and crumbling of the concrete by the vibration of passing cars. Wood blocks were used to pave this track allowance.

The practice of laying the concrete slab first has since been done away with, as we had trouble with the ties in setting the rails to grade. These ties were supposed to be 6 inches in depth but in reality ran between 6 and 7 inches with the result that we had to place steel wedges under them when bringing the track to grade.

Fig. 6 shows the method of construction used in 1915 and 1916 on tracked streets where the traffic was fairly heavy. The concrete is shown as being one solid mass, the rails being suspended and the concrete poured,

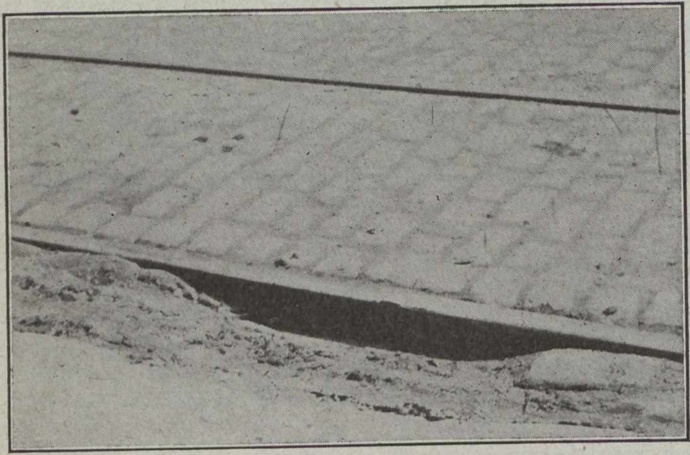


Fig. 4.—Sink Hole in Margin Blocks.

thus obviating the trouble we had as in the case of Bank Street. Sandstone blocks were used to pave the track area.

Fig. 7 shows a section of the track allowance on Rideau Street which was constructed this year. This is materially the same construction as shown in Fig. 6 with

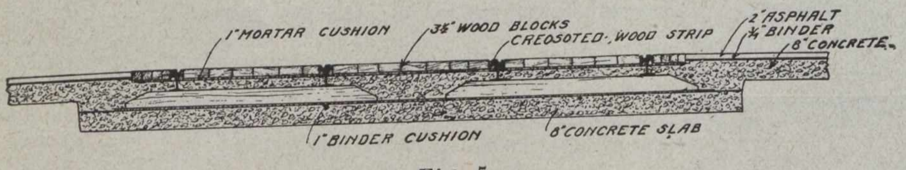


Fig. 5.

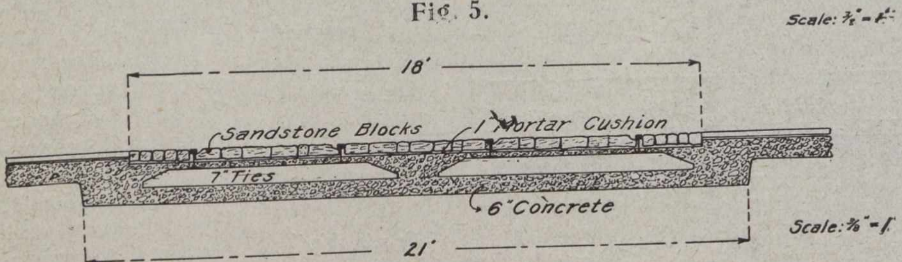


Fig. 6.

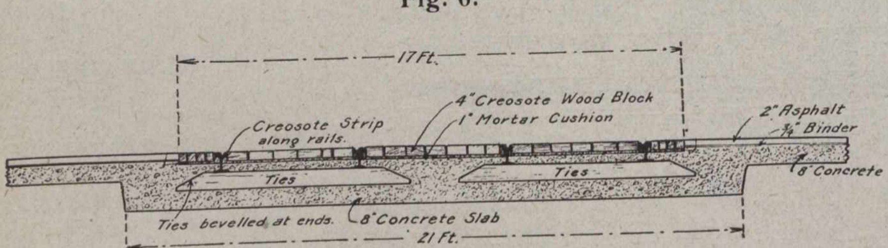


Fig. 7.

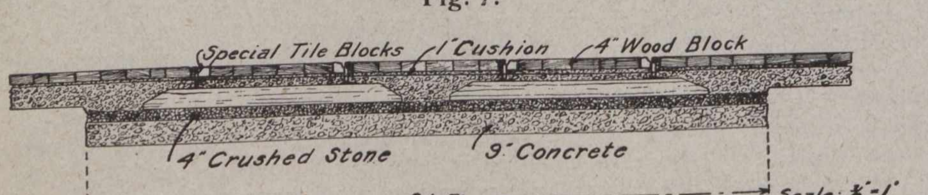


Fig. 8.

the difference that the depth of concrete under ties was increased to 8 inches and wood blocks were used instead of sandstone. An extra block was added to the margin to bring them beyond the edge of the ties, doing away with any chance of vibration which would cause the asphalt to crack.

In the Rideau Street pavement a change was made in the construction where the railway company's special work was laid. Instead of the rigid usual construction in straight track work, the concrete slab was laid first to a depth of 9 inches. On this was laid a 4-inch covering of 2-inch stone upon which the ties were laid, as shown in Fig. 8. This allows greater flexibility and also allows the special work to be renewed without disturbing the concrete slab. All pavement in the special work area will be constructed in this manner in future.

DIESEL OIL ENGINES.*

By F. Reginald Phipps, Assoc.M.Inst.C.E.

THE use of Diesel engines has of late years come prominently to the front for electric light works, sewage and water pumping, and the writer hopes that an account of their use may be welcome and instructive.

Having been instructed by his council to carry out an electric lighting scheme for the town of which I was borough engineer, the question of the most suitable prime mover to be used was one of the most important points to be decided.

The merits of steam, oil and gas engines were carefully considered. Gas engines worked by town gas were, of course, ruled out at once by the high running costs; allowing a consumption of only 20 cu. ft. of town gas per b.h.p.-hour, the cost at the local price of 3s. 1½d. per 1,000 cu. ft. would be .75d. per b.h.p.-hour. If producer gas is used, as in some towns, the economy becomes more marked, and nearly approaches that of Diesel oil engines. Thus in the waterworks which are worked by suction gas the average yearly cost per pump horse-power hour, with anthracite averaging 30s. per ton, has continued to be about .25d.

With the Diesel engines, which were decided upon and eventually installed, the cost of fuel per kilowatt-hour, with oil at 82s. per ton, is about .30d., equivalent to a price per horse-power-hour of actual work of .22d.

Even allowing for a higher efficiency of engine and dynamo over that of engine and pumps, the advantage in economy is with the oil engine.

Comparison of Cost of Fuel.—When considering these costs the price of fuel must be taken into consideration. Anthracite delivered at the station has varied in the last eight years between 26s. and 32s. per ton, while oil fuel has undergone greater fluctuations. Oil three years ago could be obtained at about 45s. per ton at station; our present contract price to August, 1916, is 82s. per ton, while the high price of 146s. 6d. has had to be paid to secure a contract for delivery in 1917. This is subject to a reduction of 50s. per ton six months after the declaration of peace.

This price is bound to drop considerably after the war, but whether it will ever reach its old low level is open to conjecture; but it is apparent that even at

*Abstract of paper read before the Institution of Municipal and County Engineers.

82s. per ton the economy in power production is most marked.

The use of suction-gas engines, however, is limited by other reasons than the question of economy; some of these are—the difficulties of direct driving, and the unsatisfactory nature of belt driving; the tarnishing of electrical instruments and other brass work from the gas fumes; the difficulty of disposal of scrubber water, and the appreciable time required in blowing up the gas before starting. In addition, it is difficult to run on a light load for any considerable period, owing to the cooling of the fuel in the producer; also, should a heavy load be applied suddenly, the speed of the engine tends to drop, as the fuel cannot be made incandescent quickly enough to supply the necessary gas.

As regards steam, its great advantage is its reliability, but unless large units are being installed, so that superheated steam and turbines can be used, the fuel costs are high in comparison with the Diesel.

For a medium-sized engine, say, up to about 150-b.h.p., the consumption of good steam coal per p.h.p.-hour under actual working conditions of electricity supply would not be less than 2½ lb. Such coal now costs here 26s. 6d. per ton at railway, which gives a cost of .35d. per p.h.p.-hour, or an equivalent of .47d. per kilowatt-hour.

Beyond the question of fuel costs, the capital expenditure must be considered. With a Diesel engine a considerable saving is made in the cost of buildings by the fact that no boiler-house, chimney shaft or economizer is required. In addition, a large quantity of water is not required for feed and condensing purposes, though a certain amount is required for jacket circulation.

Cost of Machinery.—As regards the capital cost of the machinery, it can generally be taken that the Diesel engine will cost as much as the steam engine and the necessary boiler.

The cost of two 150-b.h.p. Diesel engines, and all requisite accessories, erected in 1914, was £2,762; the water cooler for dealing with 36 gallons per minute was £195, and the softener, of 200 gallons per hour-size, was £63.

The cost of these engines was very low, and the cost of an exactly similar engine which was added in April, 1915, was £1,739.

After carefully weighing up all the various points, the writer decided to recommend the adoption of Diesel engines, and two sets of 150-b.h.p. each, coupled to direct-current dynamos of 100 kilowatts, were installed.

The following fuel guarantees were given with the tender, and have been easily fulfilled on the test and in actual working:—

Full load	.44 lb. per b.h.p.-hour	.68 lb. per kw.-hour.
¾ "	.47 " "	.70 " "
½ "	.54 " "	.83 " "
¼ "	.70 " "	1.18 " "

The engine was also guaranteed to run satisfactorily on an overload of 10 per cent. for two hours.

Before describing the practical points in connection with Diesel engines, the general theoretical principles involved may be considered.

Theoretical Principles.—The economical working of the Diesel engine is due to the fact that the heat value of the fuel is utilized to a greater extent than in any other prime mover.

Thus the Diesel engines consume per b.h.p.-hour about 7,500 B.t.u.

Gas engines consume per b.h.p.-hour about 9,000 to 12,000 B.t.u.

Steam engines consume per b.h.p.-hour about 16,000 to 28,000 B.t.u.

This high efficiency is due in the first place theoretically to the fact that the range between the high temperature of the ignited fuel and the low temperature of the exhaust is very great.

The efficiency formula for the Carnot perfect engine cycle is—

$$\frac{\text{work done}}{\text{heat expanded}} = \frac{T_1 - T_2}{T_1}$$

T_1 being the initial absolute temperature of the gas, and T_2 being the final absolute temperature of the gas.

Practically, also, the Diesel works with such high compression that the clearance space is very small, so that there is very little surface for cooling the gases when they are at their highest temperature—*i.e.*, on ignition of the fuel.

There are no losses from boiler or producer plant, and less loss in cooling or condensing water; also fuel is only being consumed during actual running, there being no stand-by losses as in steam or producer gas. In addition, any class of liquid fuel can be used, from petroleum to tar oils; that generally in use is the thick, heavy residual oil which is left after the lighter oils—petrol and petroleum—have been distilled off from the original crude oil. This fuel oil has a specific gravity of about .92-.95, and a heat value of about 18,000 B.t.u per lb. It has a high-flash point of about 200 deg. Fahr., so that there is little risk of fire in storage or use.

Owing to the present high price of fuel oil, the use of tar oils is likely to come prominently forward in the near future. These tar oils are the heavy oils (creosote and anthracene) which are obtained in the fractionation of coal tar after the lighter oils (benzol, phenol, etc.) have been distilled off. They are also a by-product from coke ovens, which it is becoming increasingly remunerative to recover. The specific gravity is from 1.0 to 1.10, and the heat value is about 15,800 B.t.u. per lb. They cannot, however, except at high loads, be used without an auxiliary, which takes the form of a pump which delivers a very small quantity of easily burning oil, such as paraffin, into the cylinder to start the ignition of the tar oil. More frequent attention to the valves is also required, as they are more liable to clog and stick up than if a cleaner fuel oil were used.

The outstanding and distinguishing feature of the Diesel engine is that ignition is not obtained by any external agency, such as sparking plugs, lamps, etc., but by high compression generating sufficient heat to ignite the fuel mixture. The majority of Diesel engines in England, and especially the smaller sizes, are single acting and of the four-cycle type.

On the first and outgoing stroke of the piston air is drawn into the cylinder through the air valve. On the second and return stroke all valves are closed and the air within the cylinder is highly compressed, usually to about 450 to 500 lb. per square inch, with an accompanying increase of temperature to about 1,000 deg. Fahr. On the third stroke the fuel valve opens, and the fuel which has been accumulated in the passages of the fuel valve is blown through a pulverizer into the cylinder by compressed air, stored at a pressure greater than that in the cylinder. This process mixes the injection air and the oil intimately, so that when it reaches the cylinder it is in the form of a spray. Directly this spray enters into the highly compressed and heated air it automatically burns, increasing in volume and driving the piston down. This, therefore, constitutes the power stroke of the cycle. It will be seen that no explosion takes place in the true sense of the word, and this type of oil engine, therefore, works much more quietly than any other type of internal combustion engine. On the fourth stroke the burnt gases are exhausted into the atmosphere, and the cycle is then repeated.

In this cycle there are, of course, two revolutions of the crankshaft for one working stroke.

In the two-cycle type of engine there is only one revolution for each working stroke. A special scavenging pump is also required for the removal of the exhaust gases, and for filling the cylinder with fresh air. On the first portion of the ingoing stroke of the piston the exhaust gases are scavenged out of the cylinder and replaced with fresh air; on the second portion of the same stroke the air is compressed. At the commencement of the second stroke the fuel oil is injected and ignited, thus constituting the working stroke; near the end of this second stroke the exhaust gases are allowed to escape, and then a fresh cycle commences.

Practical Considerations.—In the writer's opinion two of the most important points to be borne in mind in choosing a Diesel engine are to obtain as low a speed and as low a mean effective pressure (m.e.p.) as possible. The cylinder heads and pistons are subjected to such great pressures and temperatures that any reduction of these, while still obtaining the same power, is most beneficial. The continual heating and cooling induces fatigue of the metal, and the more rapid these alternations are the quicker the fatigue. A slow speed is therefore of great value in reducing the trouble often met with in high-speed oil engines of cracked heads and pistons.

The m.e.p. is obtained as follows, the actual dimensions of the engines installed being taken:—

A mechanical efficiency of 75 per cent. is usually assumed; therefore a b.h.p. of 50 is equivalent to an i.h.p. of 66⅔.

Diameter of cylinder, 13 in.
Area of piston, 132.73 sq. in.
Stroke, 20 in.
Number of revolutions per minute, 210.

Then $\frac{PLAN}{2} = 66\frac{2}{3}$.

Therefore $\frac{P \times 1.66 \times 132.73 \times 210}{2 \times 33,000} = 66\frac{2}{3}$.

Therefore P = 94¾ lb. per sq. in. = M.E.P.

The largest maker of Diesels on the Continent quoted for an engine of this size, having a speed of 195 r.p.m., with an m.e.p. of 96.1, while that quoted by probably the largest maker in England had a speed of 250 r.p.m., with a m.e.p. of 102½.

A further point requiring attention when installing Diesels is that great care is necessary in providing a sufficiently firm and massive foundation for the engine, so as to prevent trouble from vibration. A peculiarity of this vibration seems to be that it is not very noticeable in the engine-house itself, but is more pronounced some distance away. Trouble has arisen in various towns from this cause, and an injunction has been obtained against a neighboring council at the instance of an aggrieved householder living near a Diesel engine station.

In the majority of stations nothing more appears to be done than to provide a massive foundation of rich concrete carried down on to a solid bearing. In one or two instances a cushion of a few inches of sand has been interposed between the base of the concrete and the substratum. A space has also in some cases been left round the sides and ends of the concrete so that no vibration should be transmitted to the adjoining earth. Several patent methods of combating this trouble are also in vogue, comprising chiefly the use of cork layers to surround the foundation concrete.

In the writer's case a plain 5 to 1 concrete foundation was used, carried down to a depth of 7 ft. on to a solid chalk foundation, and no trouble has yet been experienced.

Nuisance is also occasionally caused by noise and fumes from the exhaust, and noise from the air-inlet pipe. A quiet exhaust can, however, generally be obtained by providing an underground baffle pit, from the end of which the vertical exhaust pipe is carried up to the roof of the building, and capped with a cast-iron silencer. A smoky exhaust should never occur if the valves are in correct adjustment, except possibly for a few moments at starting.

The air inlet pipe requires silencing, and this is usually done by closing the end of the pipe, and providing a series of long, narrow slits in lieu of the open end. If the air-inlet pipe is carried outside into the open, a quieter engine-house is the result, and the air obtained is certainly no less suitable for combustion purposes than that inside. On the other hand, in a confined engine-house the ventilation obtained by drawing air from inside the engine-house might be an advantage.

In actual running special care is required to see that the lubrication of all parts is adequate. Failure in this direction has been the cause in many cases of melting of the white metal of the bottom bearings, and in some cases this has been followed by fracture of the crankshaft—an expensive item to replace.

Other results often due to faulty lubrication are seized and cracked pistons, though the latter more often is caused by running the engine for long periods on overload and at high speeds. Many engineers on this account hesitate to put their engines for long runs on more than 90 per cent. of their rated capacity, and if this is necessary it should be taken into account in the comparison of capital costs.

Arrangement of Engines.—The general arrangement of Diesel engines is that of having one, two, three or four cylinders in a line, the power of these cylinders varying as desired. Few English makers go beyond 125 h.p. per cylinder.

The cylinders are carried in the usual "A" frame, which is in its turn bolted on to the bed-plate; the latter is secured by 2-in. holding-down bolts, going 5 ft. down into the concrete foundation, finishing with 9-in. plates, the whole as usual being grouted up with cement.

On top of the cylinders are bolted the cylinder heads, on which are fitted the fuel valve in the centre, with the starting valve in front, and the exhaust and air inlet valves on either side. These valves are actuated by three outside levers, worked from a camshaft supported on the "A" frame; in most cases the valves are driven direct from the camshaft without the intervention of levers, but it is claimed that the former has an advantage by simplifying the casting of the cylinder heads and "A" frames in doing away with the projections to carry the camshaft bearings; also that the starting gear can be more easily operated from the ground, and again that it makes it possible to enclose the whole of the cams and rollers in an oil bath, so reducing wear and noise.

The camshaft is operated from the crankshaft by a vertical driving shaft, fitted with skew and bevel gearing. The same shaft also drives the governor, which controls the speed of the engine in a very sensitive manner. Each cylinder has its own fuel pump driven by an eccentric from the camshaft, and the governor acts by limiting the amount of fuel oil delivered by this pump. It does this by controlling the suction valve. When the speed increases, a tappet underneath the valve is lifted by the governor, the period of lift varying with the speed. Thus, with a full load and normal speed, the tappet does not touch the suction valve, but if the load is lightened and the speed automatically goes up, the suction valve is proportionately held up during part or the whole of the suction and delivery strokes, thus limiting the amount of fuel oil reaching the cylinder.

The fuel pump is also provided with a by-pass, which allows the pipes to the fuel valve to be charged with fuel by means of a hand pump, so as to facilitate starting up.

The camshaft also operates a lubricating pump, which forces the oil round the cylinder walls, and also to the gudgeon pin of the piston. This oil is supplied by drip feed from an oil reservoir bolted on to the standards, another drop from the same reservoir going direct to the centrifugal oiler bolted on to the crank for lubricating the big ends of the connecting rod. The main and camshaft bearings are ring-oiled, and fitted with inspection covers for maintaining the proper oil level.

The air compressor is driven off the extreme end of the crankshaft, and supplies air to the starting and running cylinders or bottles. It is of the two-stage type, the bottom low-pressure piston compressing up to a maximum of 100 lb. per square inch, and the top high-pressure piston raising this to the blast pressure required, a maximum of about 750 lb. The compressor is water-cooled in the usual way.

The compressor delivers to two large bottles, the air from which is used for starting up the engine, and also to one small bottle for supplying the air for blowing in the fuel oil. The bottles are usually of welded steel, and are tested to 2,000 lb. per square inch. Three pressure gauges are provided, giving the pressures in the intercooler, the starting bottles, and the running bottle.

After the engine has been started up, the throttle valve on the low-pressure side is opened fully, so as to restore the full pressure to the starting bottles, and it is then adjusted to maintain the correct pressure in the running bottle.

The fly-wheel is of massive proportions, being 8 ft. 6 in. diameter and 7½ tons in weight, and is in halves. It ensures very steady running, the cyclical variation being 1/150.

The fuel oil is stored in a large cylindrical iron shell outside, holding about 23 tons. It is pumped thence by hand to the daily service tank in the engine-house, holding 200 gallons, and gravitates from this to the filter tank, which strains out any thick matter which would clog the pipes.

A second filter tank is also provided for the storage of gas oil or paraffin, and the engine is run from this for starting up, and for about five minutes before shutting down, in order to leave the valves, etc., in a cleaner condition. A three-way cock allows either oil to be used as required. The oil gravitates to the fuel pump, and thence is delivered to the fuel valve, situate on the top and in the centre of the cylinder cover. Inside the fuel valve is a pulverizer, which consists of several brass plates with small holes through which the oil is forced, thus dividing it into a fine spray. This spray of oil is forced into the cylinder by the blast of compressed air from the running bottle.

The amount of compression in the cylinder at the moment the fuel is injected is constant, and generally averages about 450 lb. per square inch, but this can be varied to suit the particular type of oil used. The running-bottle pressure varies according to the load from 550 lb. to 750 lb. per square inch.

Cooling.—A water-jacket having a 2-in. water space surrounds the cylinders, and the flow of water should be sufficient to keep the temperature of the outlet water below 140 deg. Fahr.

It is essential that the jacket space should be kept free from scale or deposit, and the circulating water must therefore be soft and clean. The town water is used in this case, and as the hardness is 15 deg., of which 11 deg. are temporary, it is softened before use. If the cooling water is supplied at a temperature of 60 deg. Fahr., and the outlet does not exceed 140 deg. Fahr., about 6 gallons per b.h.p. hour are required. Unless, therefore, suitable water can be obtained from a canal or stream, the cost of the circulating water would be a considerable item. To reduce this, a Heenan & Froude's cooler has been installed, by which the actual amount of water wasted is reduced to 2½ per cent. of the above figure; the actual amount of water required for the generation of 200,000 units was 39,000 gallons; the cooler is compact and economical, and can deal with 36 gallons per minute.

The gearing and pump are actuated by a 5-h.p. motor, and 3 units an hour are used with both engines running. At "works cost" price the current used costs 3d. per hour.

The cooler comprises a circular drum containing galvanized steel sheeting wound in a spiral, which dips for about a third of its depth into a receiver, and is revolved through gearing from the motor. The exhaust water gravitates to the receiver, and is picked up by the slowly revolving plates. A current of air is drawn through these plates by means of a fan, and the resultant evaporation of the thin film of water picked up produces a considerable cooling action by the abstraction of the necessary latent heat from the plates, in addition to the cooling due to the contact of the air and water. The cooled water is taken from the other end of the receiver and is lifted to an overhead storage tank by a centrifugal pump. This water is then ready once more to circulate through the water-jackets. The only make-up water, therefore, required is to replace that evaporated, and only this small quantity requires softening.

Dealing as we are with comparatively high speed and considerable pressures, the liability to accidents is probably greater than with steam, but, given a well-designed Diesel of adequate strength to withstand the stresses set up, these accidents should be few and not of serious character. The economy in fuel is undoubted, and ought to outweigh the attendant disadvantages.

Cost of Working.—The works since completion in October, 1914, have been under the able management of Mr. George Broadhurst, A.M.I.E.E., who, having previously acted as clerk of works during construction, was appointed electrical engineer, and so successful have they been that a third engine of similar size, and from the same makers, had to be put down last summer.

The financial results have also been most satisfactory, the following being the figures for the year ending March 31, 1916:—

Earnings		£3,212
Expenditure—Working costs	£1,725	
Loan charges	1,162	
		2,887
Net profit		£ 325

Units sold, 406,465. Load factor, 22.4 per cent.

Total costs per unit sold.

Working costs—	Fuel oil	.33d.
	Oil, waste, water and stores	.05d.
	Wages	.18d.
	Repairs and maintenance	.14d.
	Rates and taxes	.04d.
	Management, salaries, insurance, etc.	.27d.
		1.01d.
Loan charges, interest and redemption		.68d.
	Total cost	1.69d.

It is very rarely that a small town can show a net profit in the first year of an electrical undertaking, and this fact and the lowness of the working costs per unit sold are a great testimony to the efficiency of the Diesel engine. The attention of municipal engineers might usefully be turned more closely to this source of power in the future when considering the installation of water and sewage pumping machinery.

GREAT BRITAIN'S BLACKLIST

Announcement has been made by the British Government that a consolidated statutory list of enemy firms in foreign countries with whom trading is prohibited, complete to July 18th, has been issued by H.M. Stationery Office, London, England, and may be obtained by any person, post free, upon receipt of the price, six cents.

The official publication of this list makes unnecessary a continuance of the publication of the lists in *The Canadian Engineer*. The official list is the same as the lists published during the past four weeks in *The Canadian Engineer*, so far as the following countries are concerned: Brazil, Argentina, Uruguay United States, Bolivia, Chile, Colombia, Cuba, Ecuador, Paraguay and Peru.

It also contains lists of enemy firms in Japan, Netherland East Indies, Philippine Islands, Denmark, Norway, Sweden, Greece, Netherlands, Portugal, Spain, Portuguese East Africa, Morocco and Persia. In regard to the last three mentioned countries, some removals from the lists have been made, so that all firms in those countries are not now under the ban, as had formerly been announced.

Changes and corrections in the official lists will appear from time to time hereafter in the Board of Trade Journal, which is published weekly under the authority of H.M. Stationery Office, and can be purchased for six cents the copy from Wyman and Sons, Limited, 29 Bream's Buildings, Fetter Lane, E.C., London, England.

Following are the changes up to date in the lists that have already appeared in *The Canadian Engineer*:—

Additions to List.

ARGENTINA.

Bauer, P., & Company, Calle Piedras 132, Buenos Aires.

BOLIVIA.

Albrecht, C., & Company, La Paz.
Arnold & Company, Santa Cruz de la Sierre and Riveralta.
Blau, Stephen, La Paz.
Enss & Webber, La Paz.
Nolte, E., & Company, La Paz.

BRAZIL.

Andrade Pinto, Ernesto, Bahia.
Araujo & Boavista, Rua Buenos Aires 4, Rio de Janeiro.
Campos, Alexandre, & Company, Rio de Janeiro; Sao Paulo and Santos.
Companhia Sul-Americana de Electricidade, A. E. G., Rua do Hospicio 59, Rio de Janeiro.
Ferreira Bastos, Antonio, Bahia.
Fischer, Julio Christiano, Porto Alegre.
Guimares, F., Bahia.
Krahe & Company, Rua dos Andradas 497, Porto Alegre.
Linhares, Antonio P., Para.
Luckhaus & Company, Rua General Camara 67, Rio de Janeiro.
Ludwig e Irmaos, Rua dos Andradas, Porto Alegre.
Martin, Xiste, & Company, Rio de Janeiro; Sao Paulo and Santos.
Pereira, Alfredo Martins, Manaos.
Prejawa & Company, Rua da Alfandega 70, Rio de Janeiro.
Reiniger, Schmitt & Company, Rua 7 de Setembro 118, Porto Alegre.
Smith, Kessler & Panke (Casa Kosmos), Rua Direita 12, Sao Paulo and Santos.
Stoltz, Hermann, & Company, Avenida Central 66-74 (Rio Branco 66-74), Rio de Janeiro; Praca da Republica, Santos; Rua Alvares Penteado 12, Sao Paulo and Pernambuco.

CHILE.

Armstrong, Enrique, Talcahuano.
Chassin Trubert, Julio, Concepcion.
Escobar, Jose Ignacio, Calle Santa Domingo 1372, Santiago.
Guttmann & Maurer, Correa Casilla 85 and Calle Moneda 1065, Santiago; and Valdivia.
Inojosa, Maximo, Concepcion.
Koster & Wyneken, Calle Lincoyan 427, Concepcion; and Coronel.
Neckelmann & Company, Valparaiso.
Nissen, Fischer & Company, Santiago and Concepcion.
Sociedad Imprenta y Litografia Universo, Santiago.
Vargas, Leonidas, Antofagasta.

ECUADOR.

Orenstein & Koppel.

PERU.

Arce, Don Jose Elises, (of Emmel Hermanos), Arequipa.
Bast, Rodolfo, Piura.
Gildemeister, Enrique, (of Gildemeister & Company).
Weiss, Carlos, & Company, San Pedro 111, Lima; and Callao.

URUGUAY.

Castillo, Geraldo, Montevideo.

Removals from List.

BRAZIL.

Carioca, Manoel Vicente, Manaos.
Diaz Garcia & Company, Rua General Camara 39/43, Rio de Janeiro.
Weigandt, Para.

UNITED STATES OF AMERICA.

Kupper, Hermann C., 52 Murray Street, and 536 West 111th Street, New York.

Variations in List.

ARGENTINA AND URUGUAY.

Hirsch, Alfredo, (of Sociedad Financiera e Industrial Sud Americana).
Oster, Jorge, (of Sociedad Financiera e Industrial Sud Americana).

ECUADOR.

The name of the firm with which trading is prohibited by the Order of the 18th July, 1916, is Cassinelli and Company, Guayaquil. The firm of Cassinelli Hermanos y Compania of Malecon 1811, 1812 and 1813, Guayaquil, has not been placed on the Statutory List and trading with that firm is not prohibited.

UNITED STATES OF AMERICA.

National Zinc Company, 2 Stone Street, New York.

JULY COBALT ORE SHIPMENTS.

The following are the shipments for month ended July 31st:—

	Tons.
Beaver Consolidated Mines	23
Buffalo Mines	70.5
Coniagas Mines	70
Dominion Reduction Company	246
La Rose Mine	135
McKinley-Darragh-Savage Mines	201
Mining Corporation of Canada	107
Nipissing Mining Company	131.9
O'Brien Mine	31.7
Temiskaming Mining Company	60.5
Trethewey Mine	41.9
Total	1,120
From New Liskeard—	
Casey Cobalt Mine	30.5
From Temagami—Pyrites Ore	
Rand Syndicate Company	155.8
Alexo Mine	671.1

In the article on Camp Borden by W. A. Young in the July 20th issue of *The Canadian Engineer*, reference was made to Gunite coating on the walls and roofs of the buildings. *The Canadian Engineer* is informed by Carl Weber, president of the Cement-Gun Construction Co., of Chicago, Ill., which is a contracting company that owns a number of cement-guns but which is not connected in any way with the Cement-Gun Co., Inc., of Allentown, Pa., that the work was done by his concern under his supervision, and that the Chicago company's patent Gun-crete siding and roof was used on all the structures.

THE ECONOMICAL SECTION FOR SHORT SPAN, REINFORCED ARCHES CARRYING LIGHT HIGHWAY LOADINGS.*

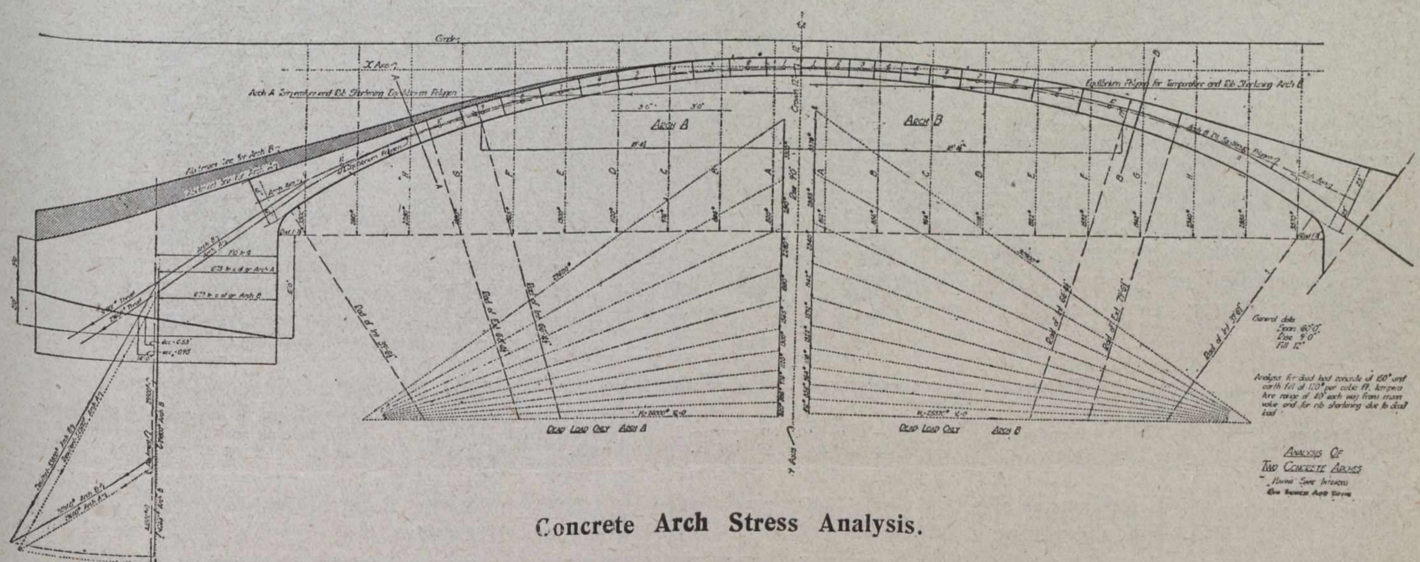
By C. B. McCullough,
Assistant Highway Engineer, Iowa Highway Commission.

UNTIL quite recent years, when the development of highway engineering has directed the attention of engineers and designers to the possibility of concrete and reinforced concrete for short span highway bridge construction, careful analytical designing, particularly of arch structures, has been largely limited to those wherein the live loadings were relatively heavy and thus a governing factor in the design. We find a great deal of literature available concerning the "linear arch" for different assumptions of loading (notably the Parabola, the Transformed Cantenary and the "Geostatic" curve of Rankine), or concerning the enveloping curves of an assumed middle third area of which the method of Alexander and Thompson is typical. The apparent purpose of all investigations of this type has been to reduce dead and to some extent live load stresses by a proper selection of the axial curve of the arch. For ordinary loadings such information is of great value to the designer. In the design of the short span, spandrel filled arch, for the lighter highway loadings, however, other factors entering into the design (particularly in localities having climatic conditions such as obtain in the northern middle west) render the selection of the depth of ring section and its method of varying from crown to spring line of equal if not greater importance than that of the axial curve.

It is a fact well known to all designers that the moments induced in a fixed arch by internal temperature changes will vary as the moment of inertia of the section or very nearly as the cube of the depth of section. (This relationship is absolutely true for plain sections and very approximately so for reinforced sections.) The moment of resistance of these same sections varies as the second power of the depth of section (strictly true for plain and approximately so for reinforced sections). Obviously, then, the fibre stress induced by the above forces will vary almost directly as the depth of section. (This relationship is somewhat modified by the direct stress, the effect of which is to increase the fibre compression and decrease the fibre tension for a rise in temperature and to modify in a reverse order for a temperature drop, but the effect of such direct stress is not large.)

While the above relationships hold equally well for structures designed for heavy live loading it is usually the case that the decrease in dead and live load fibre stress always incident to an increase in section will be sufficient to more than compensate for the increase due to temperature stresses as above set forth.

In the design of light short-span highway arches carrying a spandrel fill ordinary live loading, on the other hand, is very rarely sufficient in amount to materially modify the fibre stresses or the position of the curve of equilibrium. The dead loadings are generally much larger than the live, but do not in many cases induce the determining fibre stresses. Exhaustive experiments to determine the annual internal temperature range in concrete structures typical of those under discussion have recently disclosed a yearly range considerably in excess of that which was formerly generally believed to exist, the effect of which range is to induce stresses which in many cases are the governing factors in the design, and (paradoxical as it may seem), it is nevertheless a fact that in many cases an increase in material in the arch ring proper actually operates to raise the unit fibre stresses due to the combined loadings above mentioned.



Concrete Arch Stress Analysis.

It is not the purpose of this article to enter into any mathematical discussion of the above relationships, but rather to submit a comparison of two arch rings illustrating in a vivid manner the extent to which the above facts determine the selection of the economical shape and size of the arch ring.

The two arch rings hereinafter designated as "Ring A" and "Ring B" were recently submitted to the writer for approval. Analysis was made of each ring and the difference in fibre stresses being so noticeable, the intradosal curves were slightly modified so as to exactly superimpose and a comparative analysis made.

The crown depth for both spans is 12 in., varying through a depth of 13 in. at the quarter to a depth of 32 in. at the spring for "Ring A," and varying through a depth of 17 in. at the quarter to a depth of 46 in. at the spring for "Ring B." The comparative analysis has been made for dead load, temperature and

*Engineering and Contracting.

rib shortening, the live load being too small to materially change the stress ratios or increase the fibre stresses. Stresses were computed at the critical points along the ring and a tabulated comparison given in Table I.

A study of the table, together with the stress sheets of Fig. 1, will reveal the following significant facts:—

(1) Arch ring "B" contains about 21 per cent. more material than arch ring "A," thus increasing the cost for materials and for mixing and placing of concrete in about that ratio.

(2) Notwithstanding the above increase in material the average compression in arch ring "B" is about 11½ per cent. and the average tension about 90 per cent. in excess of the corresponding values in arch ring "A" for the same loadings.

(3) The deflection under dead load stress at the crown is .0468 in. for arch ring "A" and .037 in. for arch ring "B." Both these movements being comparatively small, the increased rigidity of arch ring "B" is but a small advantage.

(4) The abutment toe pressure for arch ring "B" is 14 per cent. in excess of that for arch ring "A" due to the greater value and less inclination of thrust at the springing lines.

$$\left[\int_0^{\frac{Z}{2}} Y\frac{ds}{I}\right] \div \left[\int_0^{\frac{Z}{2}} \frac{ds}{I}\right] \text{ where}$$

"ds" is an infinitesimal increment of axis length, "I" the moment of inertia at any point, and "y" the ordinate to the same measured from a straight horizontal line through the crown.

It is thus seen that the comparison, both as regards cost and stress condition, is favorable to arch ring "A" wherein the depth of section is not materially increased for the central half of the span. It will be noted that the equilibrium polygon for temperature is a straight horizontal line whose distance below the crown is a fixed quantity for a given arch.[1] Obviously, then, the temperature stress will decrease from crown towards spring line, passing through a zero value at the intersection of the equilibrium polygon with the arch axis and assuming its original or crown value (with opposite sign) at a point symmetrical with the crown with reference to the equilibrium polygon as an axis of symmetry. This latter point may be termed the "equivalent crown point for temperature stress." It is thus apparent that as a provision against temperature no increase in section is needed between the true and equivalent crown points.

Inasmuch as the equilibrium polygon can, by a proper selection of the central portion of the ring, be made to follow the arch axis between these two points, the dead load bending stress may be practically eliminated. The only remaining loading to be taken care of is the live and it can easily be shown that for highway loadings the increase in live load moments between the true and equivalent crown points is so small as to be negligible.

Based not only upon the example given herewith, but also upon the results of a large number of arch rings analyzed in the offices of the Iowa State Highway Commission, we are warranted in the conclusion that—

The economical section for short-span highway arch structures of which the span under discussion is typical is obtained when the axial curve is so chosen that the equilibrium polygon for dead loading is practically coincident therewith and when the depth of section between the true and equivalent temperature crown point as defined above is not materially increased.

[1]Note.—This distance, as can easily be shown mathematically, is dependent upon the elastic properties of the ring, and is measured by the quantity.

Table I.—Comparative Stress Values in Arch Rings A and B.

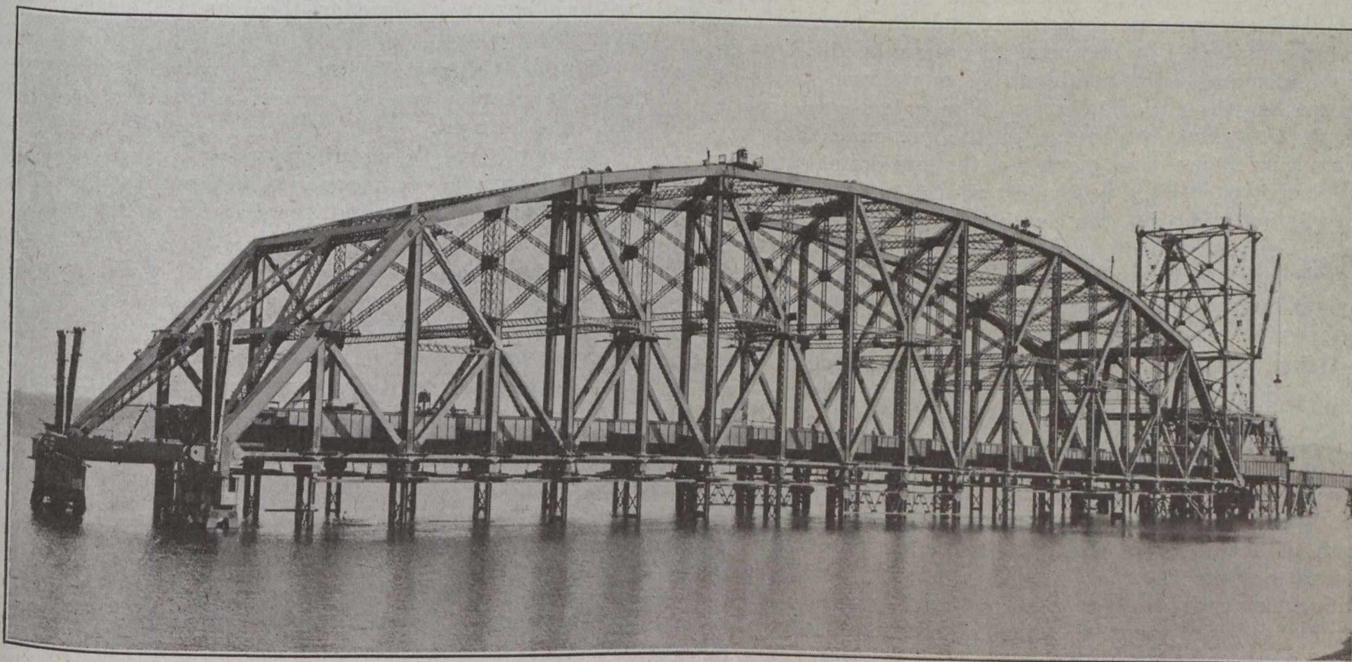
Section.	Loading.	Moment Ring A.	Moment Ring B.	Thrust Ring A.	Thrust Ring B.	*Tension in reinforcing steel. Ring A.	*Tension in reinforcing steel. Ring B.	*Compression in concrete. Ring A.	*Compression in concrete. Ring B.
Crown	Dead load	— 2,826	— 2,416	24,000	25,370				
	Temperature rise	— 5,540	— 8,108	3,600	8,123				
	Rib shortening	+ 1,385	+ 1,824	— 900	— 1,828				
	Total	— 6,981	— 8,700	26,700	31,665	785	1,300	418	521
Section 10	Dead load	+ 2,436	+ 3,496	25,500	26,220				
	Temperature rise	+ 6,260	+ 9,292	3,400	7,900				
	Rib shortening	— 1,565	— 2,095	— 850	— 1,782				
	Total	+ 7,125	+ 10,693	28,050	32,338	112	401	373	371
Section A.A. and Section B. B. (see Fig. 1)	Dead load	+ 560	+ 2,796	25,500	26,200				
	Temperature drop	— 7,710	—12,616	— 3,340	— 7,820				
	Rib shortening	— 1,927	— 2,840	— 835	— 1,760				
	Total	— 9,078	—12,660	21,325	16,640	2,750	6,070	396	386
Point 11	Dead load	— 4,711	— 2,476	27,750	27,650				
	Temperature drop	—13,080	—29,292	— 3,270	— 7,560				
	Rib shortening	— 3,270	— 6,590	— 818	— 1,700				
	Total	—21,061	—38,358	23,662	18,390	9,050	23,000	506	655
Spring line	Dead load	—23,961	—30,454	29,500	30,900				
	Temperature drop	—21,060	—49,172	— 3,240	— 7,460				
	Rib shortening	— 5,265	—11,080	— 810	— 1,680				
	Total	—50,286	—90,706	25,450	21,760	19,860	31,000	603	631

*The stresses in the material were figured on the basis of a "cracked section"—that is, the steel assumed to take the entire tensile stress. In reality such a cracked condition would alter the position of the neutral axis and therefore the pressure curve. However, for our present purpose this method gives a very good indication of the actual comparative stress conditions.

FLOATING OF CENTRAL SPAN, QUEBEC BRIDGE.

FROM information now at hand, it is expected that the operation of floating in the central span of the new Quebec Bridge will take place during the second week of September. All the main members of the structure have been completely erected and work is now being rushed on the installation of the mechanical equipment required for floating in this span.

The central, or suspended span, is a structure 640 ft. long, 110 ft. high at the centre, 88 ft. wide, and weighs over 5,000 tons lifting weight. It has been erected at Sillery Cove, some three miles below the site of the bridge and will be floated in place on six large pontoons specially constructed, three pontoons at each end. When directly under the bridge it will be connected by means of pins to long steel links suspended from the four corners of the cantilever arms. This operation will be performed at extreme flood tide, when the current will be at its minimum. As the tide recedes the pontoons will fall away from the superstructure leaving it suspended. Eight 1,000-ton jacks are then brought into play and the span is slowly lifted to its proper elevation and coupled up. The whole operation, if everything works smoothly, should not take over twenty-four hours.



View showing suspended span erected on the shore at Sillery Cove, three miles below the bridge site. Six pontoons will be floated in under this span, three at each end, and at the proper time floated to the bridge site and hoisted to its proper place.

The actual day on which the floating will take place is a question that cannot be decided until the day itself. At this point of the river there is a six to eight-mile current and a twelve to sixteen-foot tide. These features in themselves materially increase the difficulties of floating, and it will be absolutely necessary that it be a quiet day with practically a complete absence of wind.

Observations of weather have been carried on with considerable elaboration during the past couple of years, and records have been prepared showing probabilities in this respect. A wireless storm indicator has been installed at the bridge site, and records of this have been taken for the past year. This indicator will show the approach of an electrical storm some six hours before it arrives. Communication will also be kept up with the meteorological observatories in Toronto and Quebec, and all possible data will be available as soon as it can be obtained. If probabilities show that the day fixed upon for the floating of the span will be windy or stormy, the operation will be postponed until indications show favorable weather conditions.

The high tides about the period of proposed floating are between four and five o'clock in the morning. This early hour is chosen in order to give a long day for the operation.

Excursion boats and all traffic in the river will be stopped from approaching within a certain distance of the work, in order that the operations of moving this span may not be hampered in any way.

More definite information as regards data and facilities for viewing this work will be published at a later date.

COBALT ORE SHIPMENTS.

The following are the shipments of ore in pounds from Cobalt Station for the week ended August 18th:—

Right of Way Mines, 87,431; McKinley-Darragh-Savage Mine, 95,649; La Rose Mines, 87,361; Dominion Reduction Company, 152,195; Nipissing Mining Company, 329,381; Mining Corporation of Canada, 278,980; total, 1,030,997 pounds, or 515.4 tons.

The following are the shipments of ore in pounds from Cobalt Station for the week ended August 25th:—

Dominion Reduction Company, 100,000; Penn Canadian Mines, 87,469; Nipissing Mining Company, 329,860. Total, 517,329 pounds, or 258.6 tons.

From New Liskeard—

Casey Cobalt Mine, 60,931 pounds.

The total shipments since January 1st, 1916, now amount to 19,912,709 pounds, or 9,956.3 tons.

President A. H. Smith, of the New York Central, has announced that Dr. Plimmon H. Dudley, head of the railroad staff of scientists, has discovered the cause and an absolute remedy for hidden flaws in steel rails.

THE ALIGNMENT DIAGRAM APPLIED TO THE FLOW OF WATER IN UNIFORM AND COMPOUND MAINS.*

By D. Halton Thomson, M.A.(Cantab.), Assoc.M.I.C.E.

THE object of this paper is to draw attention to a comparatively new method of graphic representation, known as the alignment diagram, which, as applied to flow in water mains, has some important advantages, both in construction and use, over those now commonly employed. Not only can the usual relation between the discharge, diameter and loss of head of a single uniform main be ascertained, but the method may without complexity be extended to determine the flow in compound mains—*i.e.*, combinations of uniform mains of different lengths and diameters. The extension is made possible by taking advantage of the principle that any combination of mains of given lengths and diameters may be replaced by a single imaginary equivalent main of uniform diameter having the same discharge and loss of head. This convenient artifice enables a comparison on a simple basis to be made between various arrangements of mains and renders easier a decision as to their relative suitability in any given circumstances.

General Description.—The diagram reproduced (Fig. 1) is based on the general formula for flow in pipes:—

$$i = \frac{h}{l} = k\frac{q^m}{d^n} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

where q denotes the rate of discharge, d the diameter, i the hydraulic gradient, h the loss of head, and l the length of any uniform main; k, m, n are constants depending on the internal condition of the pipes, k in addition being dependent on the units employed. Theoretically, $m = 2$ and $n = 5$, but experiments show that the true values are in many cases slightly different. In the diagram under discussion the constants adopted are $m = 1.95$; $n = 5.068$, and if foot-second units are employed, $k = .000694$. If i is in feet per 1,000, q in millions of gallons per twenty-four hours and d in inches, then $k = 684{,}000$, m and n being unaltered, since they are independent of units. These constants are based on those given in Unwin's "Hydraulics," and apply to uncoated cast-iron pipes, the internal condition of which causes, for any given rate of discharge, a loss of head about 30 per cent. greater than that in new pipes.

In what follows the constants will be expressed in general notation in order that other values to suit different internal conditions may be introduced as desired. It is assumed that secondary losses due to changes in velocity, bends, etc., are negligible compared with the frictional loss due to the length of the pipes.

The alignment diagram consists of six vertical parallel axes with figures scales, five of which represent the variables of pipe flow given in equation (1). Where there are two scales on one axis, either may be used, according to the units in which it is more convenient to work. The remaining axis, that of "carrying power," the precise meaning and use of which will appear in due course, is introduced by the author for the purpose of calculating the flow in compound mains. Now, the fundamental property of this diagram is that the law of the scales and their relative magnitude and position are such that when any straight line, called the "index line," is laid diagonally across them, the points of intersection on each of the scales are correlated in the manner indicated by the key-diagram. It will be noticed that there are four systems of index lines—namely, $q\ d\ i$, $q\ p\ h$ or $q\ s\ h$, $i\ h\ l$, $d\ p\ l$ or $d\ s\ l$, and if any two variables of a system are known, an index line passing through their values on their respective scales will intersect the scale of the third variable at its corresponding value. It is of particular importance that the scales be always connected in accordance with the scheme shown by the key diagram. The index lines need not necessarily be drawn; a straight line scratched on the underside of a transparent celluloid strip, or a straight-edge laid in the required position, answers the same purpose. If it is desired actually to show the index lines, it is a good practice to avoid marking the diagram by drawing them on a sheet of tracing paper fixed over it.

Single Uniform Mains.—To apply the diagram to these mains, only two systems of index lines, $q\ d\ i$ and $i\ h\ l$, need be used.

Example 1.—Given discharge $(q) = 4$ cu. ft. per second, length $(l) = 4{,}000$ ft., and permissible loss of head $(h) = 10$ ft.; required, the diameter of the pipe.

Since two variables, h and l, of the $i\ h\ l$ system, are known, draw through their values on their respective scales an index line intersecting the i-scale at $(i) = 2.5$ in 1,000. Two variables, q and i, of the $q\ d\ i$ system, are now known; therefore, an index line passing through those two values on their respective scales intersects the d-axis at the required diameter $(d) = 16$ in.

Example 2.—Given discharge $(q) = 0.25$ millions of gallons per twenty-four hours, diameter $(d) = 8$ in. and length $(l) = 0.75$ miles; required, the loss of head.

Draw through the given values of q and d on their respective scales an index line intersecting the i-scale at $(i) = 6$ ft. per mile. An index line passing through the known values of i and l on their respective scales intersects the h-axis at the required loss of head $(h) = 4.5$ ft.

Carrying Power.—Now, the capacity of a main for transmitting water between two points can be defined by writing equation (1) in the form—

$$p^m = \frac{q^m}{h} = \frac{d^n}{kl} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

where p may be called the "carrying power" of that main. From this equation it is seen that "carrying power" is expressed in either of two ways:—

(1) In terms of the conditions of flow, increasing with the rate of discharge and decreasing with the loss of head, or

(2) In terms of the dimensions of the pipe, increasing with the diameter and decreasing with the length.

As will appear in due course, this conception of carrying power is of considerable utility, not only in comparing single uniform mains, but also in calculating the flow in compound mains, and for this purpose a carrying power axis is introduced into the alignment diagram. The unit adopted corresponds to a pipe discharging at the rate of 1 cu. ft. per second under a total loss of head of 1 ft.

The p-scale is connected to the rest of the diagram by means of two systems of index lines, $q\ p\ h$ and $d\ p\ l$, the use of which is best shown by examples.

Example 3.—Given a main discharging $(q) = 2.5$ millions of gallons per twenty-four hours with a loss of head $(h) = 2$ ft.; required, its carrying power.

Draw through the given values of q and h on their respective scales an index line intersecting the p-scale at the required carrying power $(p) = 3$.

Example 4.—Given main A of diameter $(d) = 12$ in. and length $(l) = 900$ ft., and main B of diameter $(d) = 30$

*Paper read at the summer meeting of the Institution of Water Engineers.

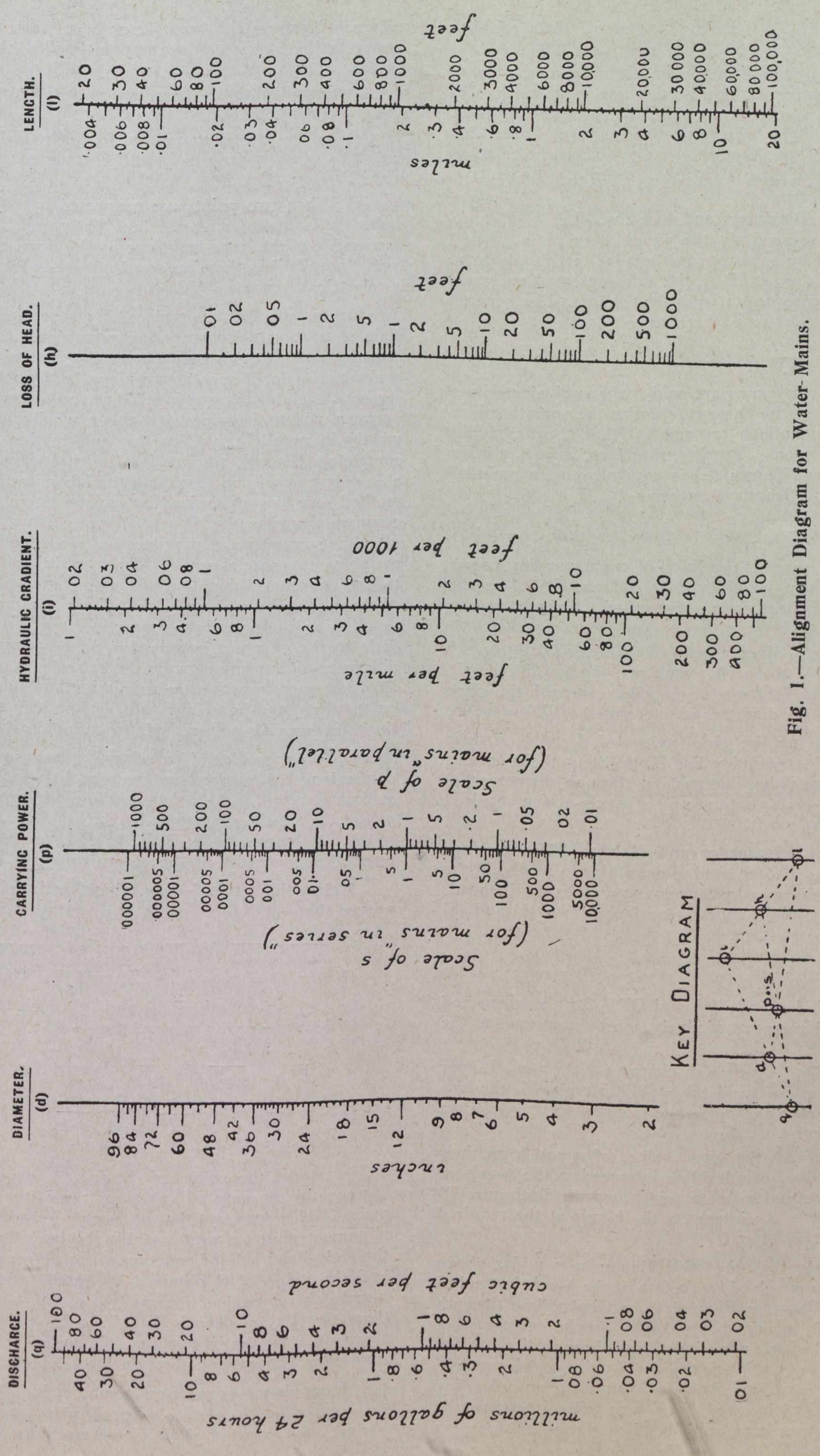


Fig. 1.—Alignment Diagram for Water Mains.

in. and length (l) = 8,000 ft.; required, their relative carrying power.

For main A draw through the given values of d and l on their respective scales an index line intersecting the p-scale at (p) = 1.2. For main B draw through the given values of d and l on their respective scales an index line intersecting the p-scale at (p) = 4.5. Then the relative carrying power of A to B is as 1.2 is to 4.5, or as 1 is to 3.75.

Equivalent Diameters and Lengths.—Let q, d, p, i, h, l represent respectively the variables of pipe flow as already defined by equations (1) and (2), and apply to a given main; let Q, D, P, I, H, L represent the same variables and apply to an equivalent main. The general equation for the latter is therefore—

$$p^m = \frac{Q^m}{H} = \frac{D^n}{kL} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (3)$$

But since two mains are said to be equivalent when they discharge at the same rate under the same head, it follows from equations (2) and (3) that their carrying power must be equal. Putting, therefore, P = p, a main diameter (D) and length (L) is equivalent to a main of diameter (d) and length (l), when—

$$\frac{D^n}{L} = \frac{d^n}{l} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

In the diagram the condition of equal carrying power is observed when the index lines dpl and DPL for the given and equivalent mains respectively intersect on the p-axis.

Example 5.—Given a main of diameter (d) = 15 in., and length (l) = 40,000 ft.; required, the length of main (L) which would discharge at the same rate with the same loss of head, if the diameter were changed to (D) = 10 in.

Draw through the given values of d and l on their respective scales an index line intersecting the p-axis at (p) = 0.3. An index line passing through diameter (D) = 10 and (p) = 0.3 intersects the l-axis at the required length (L) = 5,400 ft.

Equivalent Discharges and Heads.—The carrying-power axis may also be used to determine in any main the change in discharge consequent upon a change in head, or *vice versâ*; for it follows from equations (2) and (3) that when P = p—

$$\frac{Q^m}{H} = \frac{q^m}{h} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (5)$$

and that the index lines qph and QPH for the given and required discharges respectively must intersect on the p-axis.

Example 6.—Given a discharge (q) = 7 cu. ft. per second, and loss of head (h) = 8 ft.; required, the loss of head (H) when the discharge is changed to (Q) = 12 cu. ft. per second.

Draw through the given values of q and h on their respective scales an index line intersecting the p-axis at (p) = 2.3. An index line passing through discharge (Q) = 12 and (p) = 2.3 intersects the h-axis at the required loss of head (H) = 25 ft.

It may also be noted that Examples 1 and 2 above might have been solved by the use of the p-axis instead of the i-axis. Thus, in Example 1, an index line through (q) = 4 cu. ft. per second, and (h) = 10 ft. intersects the p-scale at (p) = 1.2; an index line through (l) = 4,000 ft., and (p) = 1.2 intersects the d-axis at the required diameter (d) = 16 in. And, in Example 2, an index line through (d) = 8 in., and (l) = 0.75 miles intersects the p-scale at (p) = 0.2; an index line through (q) = .25 millions of gallons per twenty-four hours, and (p) = 0.2 intersects the h-axis at the required loss of head (h) = 4.5 ft.

Compound Mains.—There are two cases of common occurrence:—

(1) When two points are connected by a single main with two or more lengths of different diameter laid "in series."

(2) When two points are connected by two or more mains of different diameter laid "in parallel," but not necessarily of the same length or by the same route. The term "parallel" is therefore used not in its literal, but in its electrical, sense.

Either type may be replaced by a single equivalent main of uniform diameter. Any network of mains partly in series and partly in parallel can also be replaced by an equivalent main, if such network can be divided into sub-sections, each having one inlet point and one outlet point, between which the pipes are either wholly in series or wholly in parallel. When this sub-division cannot be completely effected, the problem is in practice determinable only by trial and error, but the labor involved in this method can be minimized by expressing in terms of a common diameter the pipes of any network remaining after the system has been reduced as far as possible under the above two special cases. Moreover, it is often possible to express any combination of mains completely in terms of those cases by making slight adjustments and assumptions. For instance, cross-connections may contribute but little to the carrying power of a pipe system, since the difference of pressure between the two ends is often small; as a first approximation their omission has but little effect on the flow, and is, at any rate, on the safe side in that the total carrying power tends to be underestimated.

Let the variables of pipe-flow for each uniform component length of a compound main be distinguished by the suffices 1, 2, 3 ....; thus, the discharges by q_1, q_2, q_3, ..., the diameters by d_1, d_2, d_3 ...., and so on. As before, let Q, D, P, I, H, L represent the corresponding variables of the equivalent main.

Mains "in Series."—For these mains it is easily proved that—

$$\frac{L}{D^n} = \frac{l_1}{d_1^n} + \frac{l_2}{d_2^n} + \ldots\ldots\ldots\ldots$$

(Note: Q = q_1 = q_2 = ...and H = Σh.)

In view of equations (2) and (3), each term may be expressed as carrying power, thus—

$$\frac{1}{P^m} = \frac{1}{p_1^m} + \frac{1}{p_2^m} + \ldots\ldots\ldots,$$

$$\text{and if } S = \frac{1}{P^m},\ s_1 = \frac{1}{p_1^m},\ s_2 = \frac{1}{p_2^m}\ \ldots\ldots\ldots,$$

$$\text{then } S = s_1 + s_2 + \ldots\ldots\ldots\ldots\ldots\ldots \quad (6)$$

or, in words, the value of S for the equivalent main is equal to the sum of values of s for the component mains. This suggests that an s-scale introduced into the alignment diagram would enable the individual terms on the right-hand side of equation (6) to be read off directly and the value of S obtained by simple addition. This s-scale, therefore, has been plotted on the p-axis, and the diameter (D) and length (L) of the equivalent main are determined by the intersections on the d- and l-scales of any index line passing through the calculated value of S.

Example 7.—Given a compound main in three lengths l_1 = 5.2 miles, l_2 = 0.44 miles, l_3 = 1.5 miles, laid "in series," the corresponding diameters being d_1 = 10 in.,

$d_2 = 4$ in., $d_3 = 6$ in.; required, the diameter (D) of an equivalent uniform main of length (L) = 7.14 miles.

Draw index lines through d_1, l_1, d_2, l_2, d_3, l_3, plotted on their respective scales. At their intersection with the s-scale read off the values $s_1 = 50$, $s_2 = 500$, $s_3 = 200$, and add them together, giving S = 750. An index line passing through S and L, plotted on their respective scales, will intersect the d-axis at the equivalent diameter (D) = 6.5 (say 7) in.

Mains "in Parallel."—For these mains it is readily shown that—

$$\left(\frac{D^n}{kL}\right)^{\frac{1}{m}} = \left(\frac{d_1{}^n}{kl_1}\right)^{m} + \left(\frac{d_2{}^n}{kl_2}\right)^{\frac{1}{m}} + \ldots\ldots\ldots$$

(Note: $H = h_1 = h_2 \ldots\ldots$ and $Q = \Sigma q$.)

In view of equations (2) and (3), each term may be expressed as carrying power, thus—

$$P = p_1 + p_2 + \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (7)$$

or, in words, the value of P for the equivalent main is equal to the sum of values of p for the component mains.

This indicates that the p-scale enables the individual terms on the right-hand side of equation (7) to be read off directly and the value of P obtained by simple addition. Similarly to the previous case the diameter (D) and length (L) of the equivalent main are determined by the intersections on the d- and l-scales of any index line passing through the calculated value of P.

Example 8.—Given a compound main in two lengths $l_1 = 10{,}500$ ft., and $l_2 = 850$ ft. laid "in parallel," the corresponding diameters being $d_1 = 12$ in. and $d_2 = 8$ in.; required, the length (L) of an equivalent uniform main of diameter (D) = 15 in.

Draw index lines through d_1 l_1, d_2 l_2, plotted on their respective scales. At their intersection with the p-scale read off the values $p_1 = 0.35$, $p_2 = 0.45$, and add them together, giving P = 0.8. An index line passing through P and D plotted on their respective scales will intersect the l-axis at the equivalent length (L) = 6,000 ft.

(Continued in next week's issue.)

WEEKLY RAILWAY EARNINGS.

The following are the earnings of Canada's transcontinental railways during the three weeks ended August 21st:—

Canadian Pacific Railway.

	1916.	1915.	
August 7	$2,985,000	$1,787,000	+ $1,198,000
August 14	2,943,000	1,815,000	+ 1,128,000
August 21	2,860,000	1,956,000	+ 904,000

Grand Trunk Railway.

August 7	$1,256,376	$ 993,773	+ $ 262,603
August 14	1,236,989	1,004,412	+ 232,577
August 21	1,304,848	1,052,483	+ 252,365

Canadian Northern Railway.

August 7	$ 868,000	$ 438,500	+ $ 429,500
August 14	841,500	427,600	+ 413,900
August 21	846,300	465,400	+ 380,900

Copper and zinc to the value of $2,257,254 was sent from the Kootenay, B.C., district in the first half of 1916, against a total of $797,392 in 1915.

Twenty leading copper companies operating in the United States, Canada and South America produced approximately 895,000,000 pounds of copper in the first half of 1916, an increase of 299,000,000 pounds, or 50 per cent., over the corresponding period of last year. Of these, the Anaconda made the largest individual increase—52,300,000 pounds more than a year ago.

GRAPHICAL ANALYSIS OF CONTINUOUS BEAMS BY THE USE OF THE PRINCIPLE OF CONTINUITY.*

By Cyril Provo Hubert.

THE following analysis is general, being applicable to continuous beams having any number of equal or unequal spans.

The first step that it requires is to divide each span into three equal parts, erecting verticals at the division points (Fig. 1) then proceed to divide into half the distance between each pair of verticals over the inner supports by means of another vertical line, establishing

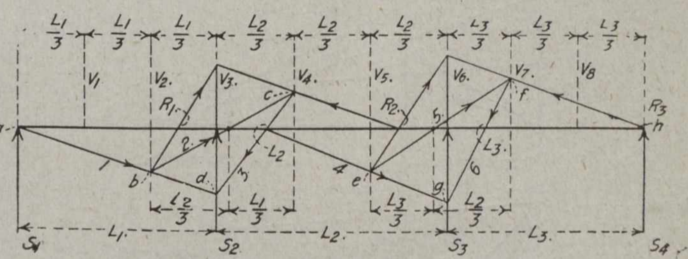
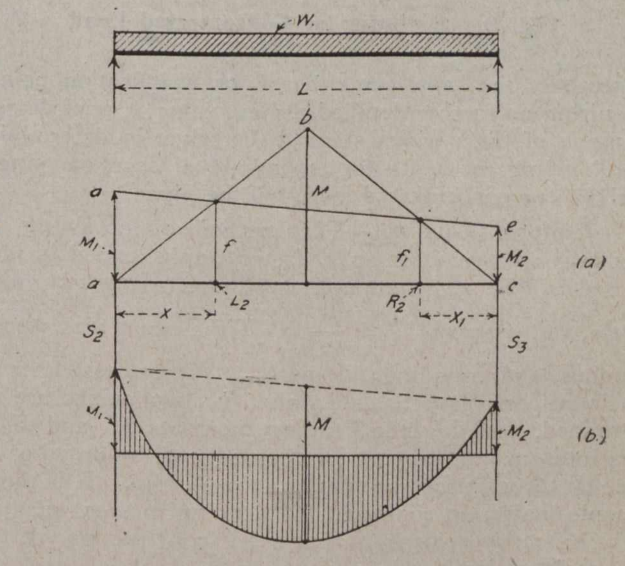


Fig. 1.—General Stress-Diagram.

the effective one-third span distances in a reverse or contrary order. It remains, then, to ascertain the position of the fixed or inflection points of each span, which must not be misconstrued for the points of contra-flexure. In order to do this by the graphical process, it is only necessary in the beginning to assume any convenient direction of the arbitrary line (1) from the point a. Where this line cuts the vertical V_2 at b, is the starting point of



$M = \frac{1}{8} WL^2$

Proof: $f = M \cdot \frac{2X}{L}$ $\quad f_1 = M \cdot \frac{2X_1}{L}$ $\quad M_1 = f + (f - f_1)\,\frac{X}{L-X-X_1} =$

$2M\,\frac{X}{L} \cdot \frac{L-2X_1}{L-X-X}$ $\quad M_2 = 2M \cdot \frac{X_1}{L} \cdot \frac{L-2X}{L-X-X_1}$

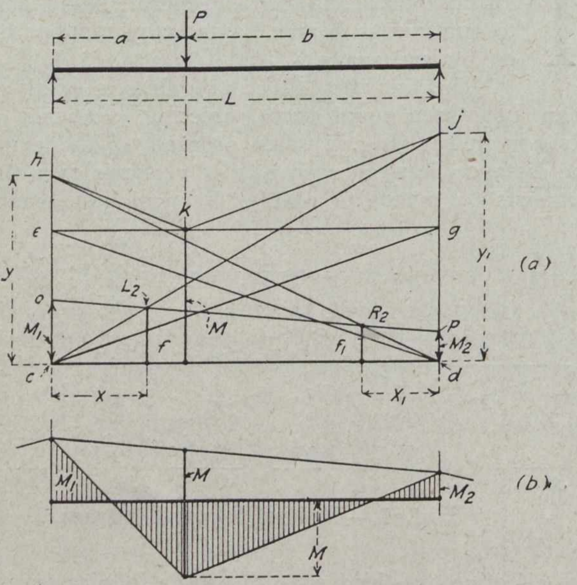
Fig. 2.—Diagram for Uniform Loading.

line (2), a straight line drawn from the point b through V_3 as shown, intersecting V_4 at c. Join the points c and d by line (3), then the left-hand inflection point L_2 of span L_2 is where line (3) cuts the datum or horizontal base line ah.

*Western Engineering.

To locate the inflection point L_3 for span L_3, start again, from the inflection point L_2 just found, and draw an arbitrary line (4), proceeding as before.

The inflection points R_1 and R_2 are obtained in the same manner, except that it becomes necessary to start from the point h, advancing to a. It is clear, therefore, that the designations regarding the inflection points are



$$M = P\,\frac{a(L-a)}{L}$$

$$\text{Proof -}\ \ y = M\cdot\frac{L+a}{L} \qquad y' = M\cdot\frac{2L-a}{L} \qquad f = y'\cdot\frac{x}{L} = M\cdot\frac{2L-a}{L}\cdot\frac{x}{L}$$

$$f_1 = y\cdot\frac{x_1}{L} = M\cdot\frac{L-a}{L}\cdot\frac{x_1}{L} \qquad M_1 = f+(f-f_1)\cdot\frac{x}{L-x-x_1} =$$

$$M\cdot\frac{2L-a}{L}\cdot\frac{x}{L} + M\cdot\left(\frac{2L-a}{L}\cdot\frac{x}{L} - \frac{L+x}{L}\cdot\frac{x_1}{L}\right)\cdot\frac{x_1}{L-x-x_1} =$$

$$M\cdot\frac{x}{L}\cdot\frac{2L-a-3x_1}{L-x-x_1} = P\cdot\frac{a(L-a)}{L}\cdot\frac{x}{L}\cdot\frac{2L-a-3x_1}{L-x-x_1}$$

$$M_2 = P\cdot\frac{a(L-a)}{L}\cdot\frac{x_1}{L}\cdot\frac{L+a-3x}{L-x-x_1}$$

Fig. 3.—Diagram for Concentrated Load.

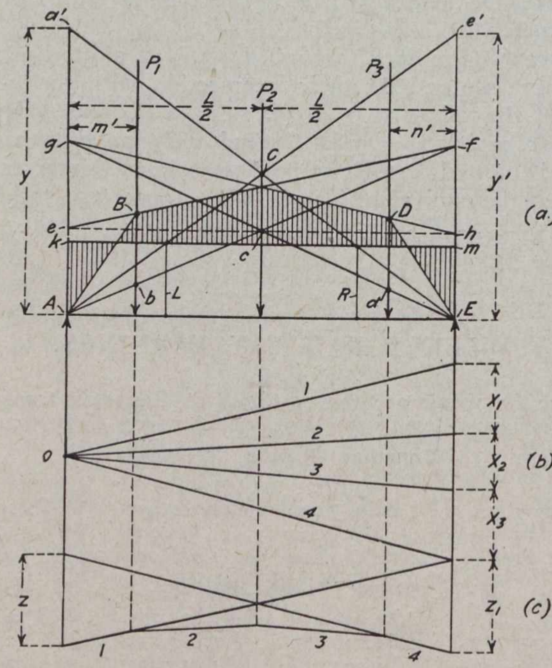
essentially the same for either L or R inflection points, requiring no further detailed explanation. A brief demonstration of the various steps to be taken in determining the bending moments for an individual span carrying a uniform or concentrated load, will be given.

Uniform Loading.—The method of obtaining the bending moments for span L_2 uniformly loaded is illustrated in Fig. 2. Compute, then, plot to any convenient scale, the moment, M at $\frac{L}{2}$, as if the beam was discontinuous, and draw lines ab and bc. Where these lines cut their respective verticals L_2 and R_2, two points are determined, enabling the bending moments M_1 and M_2 at the intermediate supports, to be deduced, by drawing the line de through these points as shown in (a). It is then a simple matter to construct the bending moment diagram for the entire span, requiring the construction of the parabola as in (b).

Concentrated Loads.—Referring to the same span L_2, as discontinuous, the effect of a concentrated load P, will now be taken into consideration (Fig. 3), M in this case being used to indicate the bending moment directly at the point of application of the load. Having plotted the value of M in (a), draw eg parallel to the base cd. Join cg and ed, and then draw hk parallel to ed and kJ parallel to cg. Join cJ and dh and where these lines cut their respective verticals L_2 and R_2, draw the line op. The negative bending moments M_1 and M_2 can then be scaled off, and the bending moment diagram (b) constructed.

Irregular Load Systems.—In Fig. 4, a system of concentrated or superimposed loads, P_1, P_2 and P_3 is considered. In order to apply the graphical method to this case, the bending moment diagram $ABCDE$ for the span, irrespective of the beams being continuous, is plotted. After this has been done, a special construction is required in order to find the positive and negative moments due to the continuity of the beam. The method is as follows: Produce line BC in both directions, until it intersects the continuation lines of the supports at e and f. By a similar reasoning, using line CD, points g and h are determined. Now connect g and E, A, and f and E and h by means of straight lines. These operations serve for determining the three points b, c, and d. Symbolizing the distances Bb, Cc, and Dd by X_1, X_2 and X_3, respectively, plot them along the line of one of the supports and with a pole distance equal to the span, draw the polar diagram (b) from which diagram (c) is constructed. Now lay off y and y' as shown in the figure and draw line $a'E$ and $A'e$. Where these lines cut the respective lines produced from the inflection points L and R, the necessary two points are obtained to determine the direction of line km, completing the diagram.

So far the bending moments for only one span have been shown diagrammatically. The moments at the supports of a continuous beam have been deduced in Figs. 5 and 6 and can be determined easily. The simplest procedure for obtaining these moments is to derive the moments for the loaded span, by the means explained above, the second element requiring recognition of the remaining inflection points, which governed the direction



$m' = n'$ $\qquad y = Aa' = (X_1 + X_2 + X_3) + Z$

$Bb = X_1$ $\qquad y' = Ee' = (X_1 + X_2 + X_3) + Z_1$

$Cc = X_2$

$Dd = X_3$

Fig. 4.—Diagram for Irregular Loads.

of the lines instrumental in obtaining these moments. To simplify explanations, the arbitrary term, instrumental line, will be introduced for want of better terminology, there being no standard technical term that can be used to designate this kind of a line. Considering the span just to the left of the loaded span, from the point of greatest bending moment (a), previously found, the instrumental line is drawn through the inflection point L_3, intersecting the support S_3, M represents the bending moment at support S_3 due to the loaded span L_4. The rest of the instrumental lines are drawn in the same manner, with the exception of the end spans, as shown. It will be noticed, from an inspection of the instrumental lines, that those to the left of the loaded span pass through L inflection points, while those to the right utilize the R inflection points.

The foregoing principles will now be applied to the case of a continuous beam with three spans, the supports being all of the same level. It is upon this assumption, namely, that the supports are all on the same level, the calculations are based, since a slight subsidence of one or another of the supports, considerably changes the bending moment at any section of the beam.

By a course of investigation similar to that indicated in Fig. 1, the inflection points are located, through the aid of which the effects of the different conditions of dead and live loading are obtained, these effects being investigated separately. Each step taken is shown in its consecutive order, the final diagram representing the bending

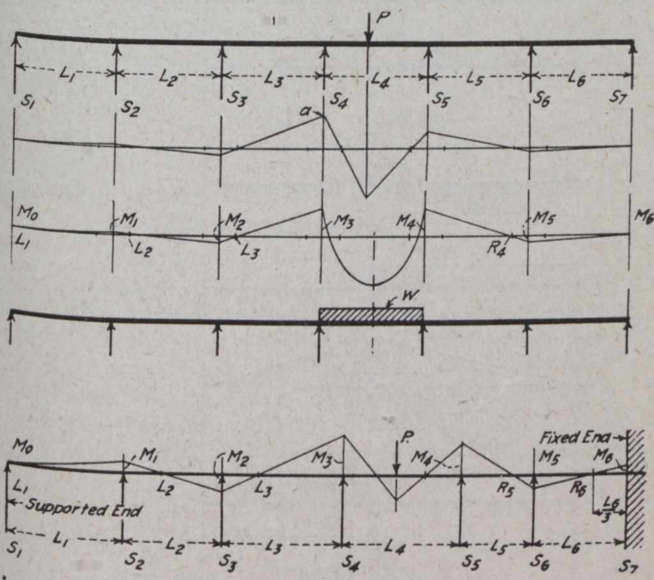


Fig. 5.—Moments in Continuous Beam Resulting from a Concentrated Load.

moment as a whole, being the outcome of superposing the various other diagrams.

Explanation of Fig. 6.—A continuous beam of three spans, the end spans being equal and the beams subjected to both dead and live, concentrated or superimposed loads.

(a) Inflection points established.
(b) Concentrated dead load on span L_1 considered.
(c) Concentrated dead load on span L_2 considered.
(d) Effect of combined concentrated dead loading of the three spans.
(e) Concentrated live load on outer spans taken into consideration.
(f) Concentrated live load on centre span taken into consideration.
(g) Diagrams (e) and (f) combined.
(h) The positive areas of (g) combined with diagram (d) upturned.
(i) The negative areas of (g) combined with diagram (d) upturned.
(j) The final diagram resulting from the combination of (h) and (i).

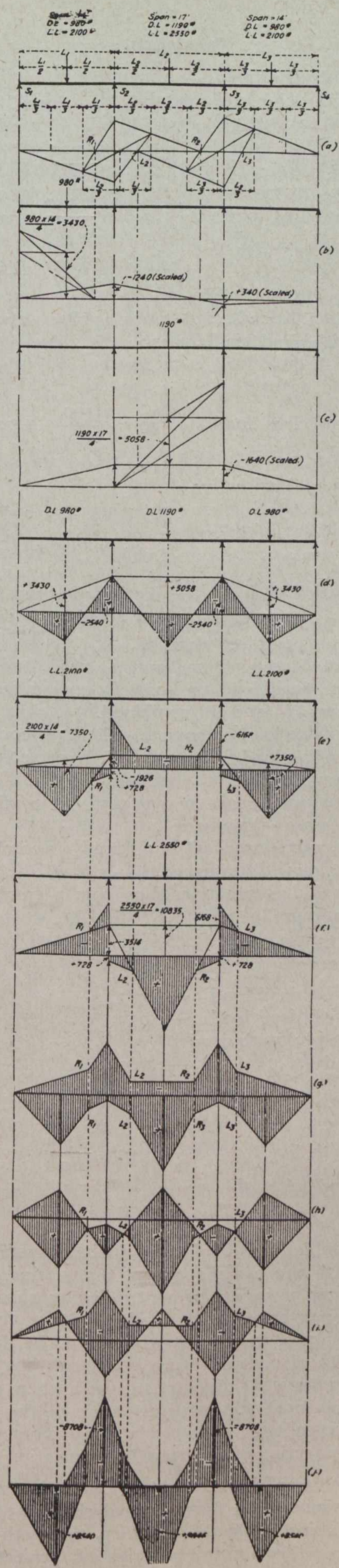


Fig. 6.—Analysis of Stress in Continuous Beam of Three Spans with End Spans Equal.

Tabulation.

Span (1) with dead load.. – 1,240
Span (2) with dead load.. – 1,640
Spans (1) and (2) with (3).. $-1{,}240 - 1{,}640 + 340 = -2{,}540$

Span (1) with live load .. $-1{,}240 \times \frac{2{,}100}{980} = -2{,}654$

Span (2) with live load .. $-1{,}640 \times \frac{2{,}550}{1{,}190} = -3{,}514$

Spans (1) and (2) with live load .... $-2{,}654 - 3{,}514 = -6{,}168$
Spans (1) and (3) with live load .... $-2{,}654 + 728 = -1{,}926$
Span (1) with dead load.. + 340
Span (2) with dead load.. – 1,640
Spans (1) and (2) with (3) – 2,540

Span (1) with live load.. $+340 \times \frac{2{,}100}{980} = +728$

Span (2) with live load .. – 3,514
Spans (1) and (2) with live load .... $+728 - 3{,}514 = -2{,}786$
Spans (1) and (3) with live load .... – 1,926

Explanation of Fig. 7.—Continuous beam of two unequal spans, carrying both dead and live uniform loads.

(*a*) Inflection points established.
(*b*) Uniform dead load on span L_1 considered.
(*c*) Uniform dead load on span L_2 considered.
(*d*) Diagrams (*b*) and (*c*) combined.
(*e*) Uniform live load on span L_1 taken into consideration.
(*f*) Uniform live load on span L_2 taken into consideration.
(*g*) Diagrams (*e*) and (*f*) combined.
(*h*) The positive areas of diagram (*g*) combined with diagram (*d*) upturned.
(*i*) The negative areas of diagram (*g*) combined with diagram (*d*) upturned.
(*j*) The final diagram resulting from a combination of diagrams (*h*) and (*i*).

Tabulation.

Span (1) with dead load..—1,725
Span (2) with dead load..—1,035
Spans (1) and (2) with dead load ....—2,760

Span (1) with live load .. $-1{,}725 \times \frac{150}{80} = -3{,}234$

Span (2) with live load .. $-1{,}035 \times \frac{150}{80} = -1{,}940$

Spans (1) and (2) with live load ....—5,174

The complete assembling of such a construction may interfere with the pet theories of many, due to the fact of there being no points of contra-flexure shown. Those thoroughly conversant with the behavior of live loads of a variable nature, know that what would be the positions of the points of contra-flexure when carrying a full live load, would be considerably stressed when only partly loaded. The most noteworthy feature of the diagram is that it represents the greatest bending moment that can occur at any section of the beam by translation of the load.

In conclusion, I wish to state that it is not my desire to give the impression that I have derived any new theory upon this subject. On the contrary, the method is based upon the combination of a series of perfectly familiar elementary formulæ, and is a matter of common knowledge in Europe, though little known here. The purpose of this

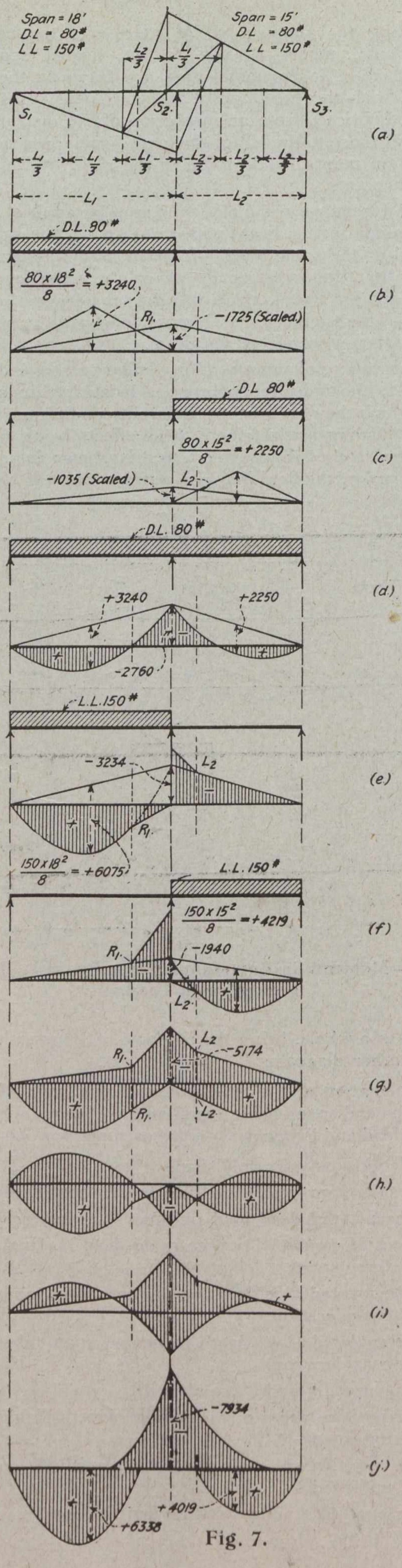


Fig. 7.

article is to show that by a careful analysis of the facts outlined, essential accuracy may be obtained without delving into laborious calculations. At present the usual method employed is either to assume the spans as discontinuous instead of extending unbroken over the supports, allowance being made for the negative moment in accordance with the judgment of the engineer, which method is based on a fallacy, or else to use the tentative formulæ now in common use with large factors of safety.

Numerous articles have been written on the merits and demerits of continuous beams and research is continued. But with all the diversity of opinion upon this subject, to the impartially minded engineer, it is evident that a structure designed in a manner that will permit of a saving of 25 to 30% of the material in comparison with other types and yet fulfil all the requirements in regard to strength and durability is to be preferred, since it is the most economical.

The employment of continuous beams is opposed in some quarters because of the danger of an unequal settlement causing material changes in the stresses. The proper way to guard against this danger, however, is by a study of the structural geology of the region, and by greater care in the design of the foundations, rather than by condemning the use of continuous beams.

PROPOSALS TO ADVANCE BRITISH ENGINEERING.

The Council of the Institution of Electrical Engineers of Great Britain has adopted resolutions stating several measures which it advocates as a means of advancement for British engineering. It is in favor of a broader recognition of high technical attainments, and among the changes it advocates is that the use of the metric system be made compulsory after a reasonable period, and that during this period all trade catalogues make use of both the British and metric systems. The resolutions, which are printed in the Journal of the Institution of Electrical Engineers in its issue of June, 1916, are:—

Some combination of British electrical firms, especially with regard to over-sea trade, is desirable.

A government tribunal of the most independent character that can be devised, to be appointed to control the electricity-supply industry of the country, and also to prevent indiscriminate addition or extension of power stations or systems undesirable from the point of view of size, locality, or system.

In view of the necessity of securing the home market and that none other than British electrical apparatus be purchased in the United Kingdom, a protective tariff to be set up, notwithstanding such benefits as will in any case result from patriotism.

A permanent advisory committee to be appointed to insure that, as far as possible, raw materials and parts as well as whole apparatus necessary to the trade of the British Empire shall be produced within the Empire.

British-born electric attachés to help in the consular service, and trade commissioners (scientific and technical commissioners) to be appointed.

British engineering standards to be adopted throughout the Empire.

The use of the metric system to be made compulsory after a reasonable period, and during this period all trade catalogues to make use of both the British and metric systems.

The institution to be granted a charter so as to improve the status and training of electrical engineers.

A central engineering board, consisting of representatives nominated by all the important institutions, to be established whom all engineers (other than mechanics) would be required to satisfy as to the sufficiency of their technical training and general education before they could be recognized as proficient, so as to insure that every engineer shall qualify for his profession in the same manner as a doctor or solicitor.

Closer co-operation of manufacturers and other employers of electrical engineers with the technical colleges is desirable to insure that students are trained to meet the future needs of the industry.

THE RARER METALS.

An interesting paper on "The Metallurgy of the Rarer Metals" was presented by Prof. J. W. Richards, of Lehigh University, South Bethlehem, before the eighth semi-annual meeting of the American Institute of Chemical Engineers. He said that in 1866 aluminium was one of the rarer metals selling at $10 per pound. The silicon industry furnished another example of great reduction in the cost of a metal, as in 1900 it was selling as a chemical curiosity at over $100 an ounce; now 10 cents per pound was a good market price for it. The speaker referred at length to the number of metals that at present command high prices, but which by improved metallurgical processes might be made very cheaply. He said the present methods of reducing beryllium were tedious and costly. This metal being white, malleable, and unchanged in air, with a specific gravity of 1.64, would be particularly useful for objects where great lightness and permanence in air were the first consideration, cost being secondary. Although magnesium oxide cost only a few cents per pound the metal sold for about as many dollars per pound, and was scarce. But it was believed that, by improving the methods, the reduction of magnesium could be effected at 25 cents per pound. After the war, under normal industrial conditions, magnesium would sell at a price which would take it out of the class of rarer metals and put it among the common ones. As the price went down its industrial uses would increase in geometrical proportion, and instead of the production being expressed in thousands of pounds per year it would reach thousands of tons. Its alloys might largely displace aluminium alloys, which were now used by thousands of tons annually in the motor-car industry, with a saving of one-third in weight which would compensate for the higher first cost. He predicted that the metallurgical use of magnesium would also be greatly extended by its lower price, such as for deoxidizing brass, copper, bronze, nickel, and monel metal, since it was a much stronger deoxidizer than aluminium. A small addition of metallic calcium might be used to reduce the amount of sulphur and phosphorus in steel, and other metals and alloys whose properties were damaged by sulphur or phosphorus might be similarly refined or improved. Chromium electroplating was white and durable, and for many purposes might be superior to nickel and almost equal to platinum plating, but the technique of always getting perfect plating had not been mastered. The metallurgy of chromium was full of attractive possibilities, and the usefulness of pure chromium in the field of alloys was only beginning to be investigated.

FINK TRUSS WEB STRESS ANALYSIS BY NEW METHOD.

By Harry B. Wrigley.*

THE graphic analysis of web stresses in Fink trusses presents, according to writers upon this subject, a difficulty at joints 4 and 5, Fig. 1. This difficulty results from three unknown forces at each of the joints named for, in a system of forces in equilibrium, acting through the same point, in the same plane, at least all conditions but two must be known in order to construct a force polygon.

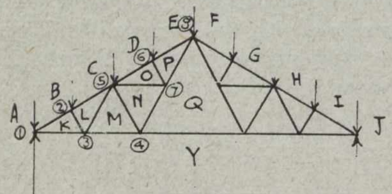


FIG 1

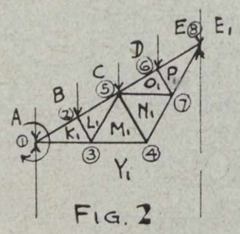


FIG. 2

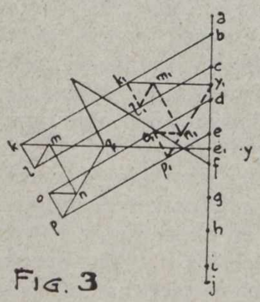


FIG. 3

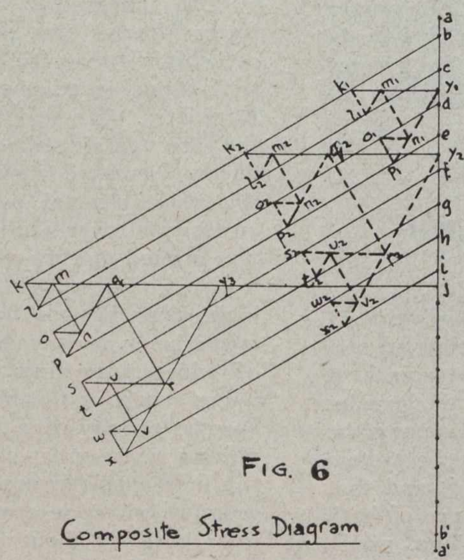


FIG. 6

Composite Stress Diagram

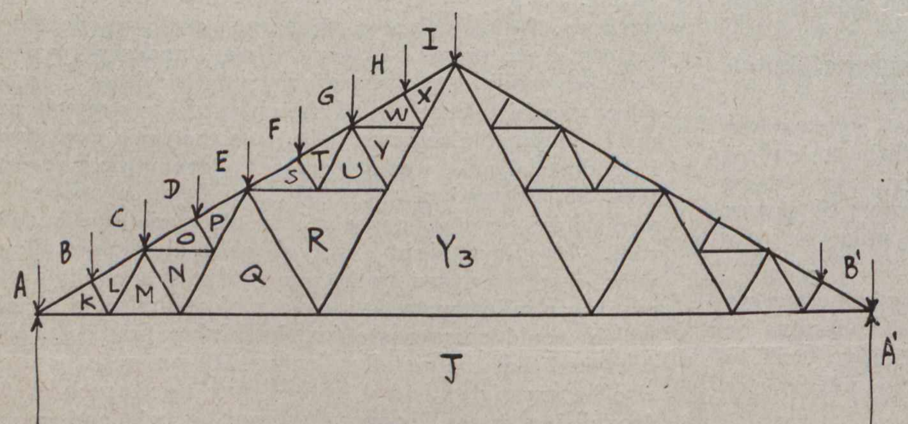


FIG. 4

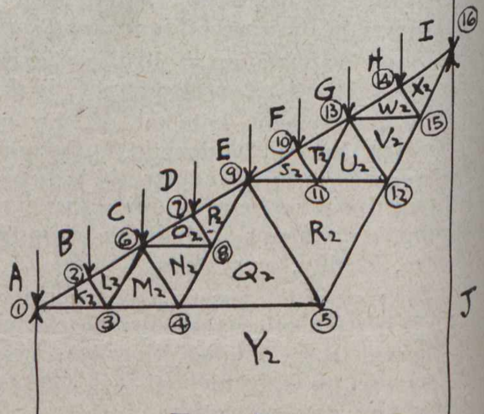


FIG. 5

In "Roof and Bridges" (Merriman and Jacoby) there is presented a solution by Willett, and described at a meeting of the Chicago Chapter, American Institute of Architects, in 1888.

Goodman, in his "Mechanics Applied to Engineering," gives a solution by Professor Barr, Glasgow University, identical with Willett's solution, requiring the substitution of an imaginary member connecting joints 4 and 6 to replace members N-O and O-P, Fig. 1.

Other writers, who have not adopted the above method, suggest a solution which fails under certain conditions of roof loading, consequently I have developed a new and logical solution to be employed regardless of roof loading conditions.

To illustrate, consider Fig. 2 (part of Fig. 1). Lay off the load line a—e_1 and reactions e_1—y_1 and y_1—a, Fig. 3, then construct the stress diagram for the web members as shown in dotted lines, taking the joints in numerical order. It is evident from Fig. 1 that panel loads to the right of E-F do not affect the web members to the left, and vice versâ. Hence, having determined the web stresses, complete the load line a—j and reactions j—y and y—a, Fig. 3, to obtain the stresses in the chord members. The web stresses in the final diagram are identical with those in the dotted diagram.

Similarly, in the compound Fink truss (Fig. 4) the web stresses are easily obtained. Keeping in mind that stress m_2—n_2 is the same as m_1—n_1, the stresses for joint 4 (Fig. 5) are indicated by the polygon y_2—m_2—n_2—q_2—y_2, Fig. 6. Having thus determined q_2—y_2, the remaining web stresses are obtained by taking the joints in numerical order, as shown, from whence the complete stress diagram is drawn, as was done in Fig. 3.

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NICKEL COMPANY'S PLANS.

The International Nickel Company will spend about $2,000,000 on the proposed new refinery which the company is to erect in Canada to produce all nickel needed by Great Britain and overseas dominions. This expenditure will be met from the company's treasury funds which approximate $8,500,000.

A subsidiary concern has been formed in Canada to own and operate the new plant, and its $5,000,000 capital stock will be owned by the International Nickel Company.

Under the agreement entered into between the company and the Canadian government officials, British nickel requirements will be made in Canada, the balance at the New Jersey refinery. It is estimated that the cost of producing refined nickel in Canada will be but little higher than in the older plant in the United States. Arrangements have already been entered into whereby the company will secure its power from the government at but little over the actual cost price.

Editorial

WAR EXPORTS AND PEACE

The total trade of Canada for the twelve months ended June, exclusive of coin and bullion, was valued at $1,565,436,000. Of this, imports of merchandise represented $595,921,000 and the exports $969,514,000. The difference between the value of our imports and exports of merchandise in that period was therefore $373,000,000. After making allowance for the payment of $187,000,000 interest charges to Great Britain and the United States, there is a balance of trade in favor of Canada amounting to $186,000,000.

Of the exports, totalling $966,514,000, only $96,000,000 represented the export of foreign produce, the remaining $873,413,000 being exports of Canadian produce. These were made up as follows:—

Mine	$ 71,834,835
Fisheries	23,248,778
Forest	53,259,354
Animal produce	108,147,108
Agricultural produce	323,510,530
Manufactures	284,495,047
Miscellaneous	8,917,802

Comparing these figures with those for the similar period of 1914, it is found that our exports of mineral products have increased during the year 22 per cent.; fisheries exports by 15 per cent.; exports of forest products, 23 per cent.; agricultural products, 75 per cent.; animal produce, 96 per cent.; agricultural produce, 75 per cent.; the exports of manufactures, by 365 per cent.; and miscellaneous classes by 4,426 per cent.

Agricultural products represent the biggest item in our exports and probably will always do so. But the fact that Canadian manufacturers have been able to increase their export trade to a volume nearly five times as great as it was two years ago, gives an idea of the productive power of Canadian factories. The present unusual demand for Canadian manufacturers' goods is due to the war. The problem which confronts them is to measure and encourage, at home and abroad, the demand for their goods after the war. Something substantial has to be found to take the place of war orders. How many manufacturers are allowing to-day's prosperity to shadow to-morrow's problems?

PLACING OF CENTRAL SPAN, QUEBEC BRIDGE.

Announcement has been made that on September 11th traffic on the St. Lawrence River at Quebec will be suspended for twenty-four hours in order to permit the central span of the Quebec Bridge to be swung into position.

This is an event that will attract the attention of engineers all over the world inasmuch as it will mean the linking together of the two ends of a structure the development of which has been watched with great interest since it was commenced a few years ago.

Public interest too in the successful completion of this work has been high because it means the placing into position of the last link in the unbroken rail to rail highway which will then extend from ocean to ocean.

The design, construction and erection of the Quebec Bridge constitutes one of the really great engineering feats of modern times, and will occupy for a long time to come an increasingly important position in the ranks of great national engineering undertakings.

This central span, an excellent picture of which appears in this issue, is 640 ft. long, 110 ft. high at the centre, 88 ft. wide and weighs approximately 5,000 tons lifting weight. The methods to be employed in lifting this span into position were described in detail in our issue of June 1st, 1916.

From time to time *The Canadian Engineer* has kept its readers informed as to the progress made in the construction of the bridge. It is a notable fact that while work was carried on from both sides, when the north shore anchor arm was completed, the main post was only 15 inches out of plumb, this being due to the unbalanced weight of the uncompleted structure. The calculations have been so close that when the centre span is in position, as it will be within a few days, the centre post of each main truss will be practically perpendicular.

To all who have been in any way connected with this notable enterprise, either in its design, construction or erection, *The Canadian Engineer* extends its sincere congratulations.

INTERNAL COMBUSTION ENGINES.

During the past few years there have been installed in Canada quite a number of Diesel and other internal combustion engines, many of them being put in by municipalities for electric lighting plants, pumping plants in connection with sewerage and waterworks. This is perhaps more true of our Western municipalities than it is of the East.

The number of uses to which these engines have been put is increasing, and it is likely that as time goes by municipalities will find these engines suitable for other kinds of work than that for which they are now employed.

In this connection we publish in this issue abstracts of a paper by Mr. F. R. Phipps, read before the Institution of Municipal and County Engineers, in which he makes some very interesting comparisons between steam, oil and gas engines.

In considering the deductions which Mr. Phipps makes in his paper one will have to take into account the local situation. For instance, where coal is cheap, as is the case in some sections of the Dominion, steam power will almost always win out. At the same time, the figures which Mr. Phipps gives will, we feel sure, be of considerable interest to municipal engineers throughout Canada, and we have much pleasure in presenting them.

THE CHANNEL TUNNEL.

For many years a tunnel under the English Channel connecting Great Britain and France has been talked of. Since the war began this much-discussed project has been revived in a serious form and it is generally accepted, in

England at least, as almost certain that the often-contemplated work will be undertaken. When it is, it will probably have the backing of both the British and French governments.

The advances in modern engineering have been such during the past few years as to render it possible to overcome the obstacles which have helped deter actual operations in this regard, and furthermore, the alliance of the two nations has been so cemented as to remove any opposition that formerly arose on the score of military precaution.

The distance across, from Dover to Calais, is approximately 21 miles. If constructed, it will be the longest subaqueous tunnel in the world and will call for the highest type of engineering ability in its design and construction.

The actual carrying out of such an enterprise may be delayed, but the feeling is growing stronger every day that it ought to be gone on with, and doubtless before many years have passed we shall see the work commenced. It is presumptuous for anyone to say what can or cannot be done. There seem to be no obstacles from an engineering point of view and in any event it is doubtful whether it is ever wise to place any limit on the ingenuity and ability of the engineer.

PERSONAL.

A. O. WOLFF has been appointed resident engineer, District 2, Lake Superior Division, C.P.R., Chapleau, Ontario.

J. C. GWILLIM, professor of mining at Queen's University, Kingston, Ont., is at Valcartier, having joined the tunnelling company.

C. J. KAVANAGH, superintendent of Toronto C.P.R. terminals, has been transferred to Montreal to take over the terminals there.

Lieut. CLEMENT SAYE, secretary-treasurer of the Northern Electric Company, has joined the 245th Battalion Canadian Grenadier Guards, now being organized in Montreal.

R. HOME SMITH, of Toronto, has been elected president of the Mexico North Western Railway, with offices at Toronto and at El Paso, Texas, vice F. S. Pearson, deceased.

R. B. ANGUS, director, C.P.R., Montreal, and Sir THOMAS TAIT, president, Fredericton and Grand Lake Railway and Coal Co., have been elected Fellows of the Royal Colonial Institute.

C. P. VanNORMAN, resident engineer, Toronto and York Radial Railway, Toronto, is attached to the 127th Battalion (York Rangers), with the rank of lieutenant, and is stationed at Camp Borden.

Sir GEORGE PAISH, who was named recently as one of the commission appointed by the Dominion Government to enquire into the general railway situation, is stated to have declined the appointment on account of ill health.

Lieut.-Col. H. N. RUTTAN, M.Can.Soc.C.E., consulting engineer for the city of Winnipeg, and formerly city engineer, who has been for some time district officer commanding at Winnipeg, has been promoted to brigadier general.

Dr. D. B. DOWLING, of the Department of Mines, Ottawa, has gone to Lethbridge, Alberta, where he will endeavor to find an artesian water supply in southeastern Alberta. Three wells will be drilled in an area examined by Dr. Dowling last year.

GEO. A. WALKEM, managing director of the Vancouver Machinery Depot, who recently left for England with the object of joining his Majesty's forces, has obtained a commission in the Royal Engineers, and has left for Egypt, where he will be stationed for the present.

J. C. COUTTES, well-known in Winnipeg through his work on the Shoal Lake pipe line, ARCHIBALD PAGETT, formerly with the C.P.R., and H. C. HAYWOOD, three well-known Winnipeg engineers, two of whom qualified as captains, have just signed up as privates with the 239th Construction Battalion.

OBITUARY.

WILLIAM WHEATLEY, contractor, Clinton, Ont., died on August 28th.

ALPHEUS BEEDE STICKNEY, who was the first general superintendent of the C.P.R. at Winnipeg, appointed on the organization of the company in 1881, died at his home in St. Paul, Minn., August 9, after an illness of four weeks.

JAMES D. POWELL, of Brockville, Ont., died recently at the age of 72. Mr. Powell was born in Brockville, and upon attaining manhood he moved to New York City, where he engaged in business as a building contractor.

ANGUS W. McISAAC, for two years city inspector of electric wiring at Sydney, N.S., is dead. For thirteen years previous to accepting the position with the city he had been in charge of the power station of the Cape Breton Electric Company.

JOHN HOLMES, one of the old residents of the east end of Toronto, died recently after a short illness. Mr. Holmes was a native of Cambridgeshire, England. He came to Canada at the age of seventeen, settling in Stratford, where for many years he was a prominent contractor and also chairman of the Board of Works. For the last 43 years he has been a well-known valuator in Toronto.

Gunner THOMAS LEON GOLDIE, of the 16th Battery, C.F.A., died from infective jaundice in the Western General Hospital, Manchester, England, on August 28th. Gunner Goldie was in his 34th year. He was a son of the late Mayor Thomas Goldie, of Guelph, Ont., and was born and received his early education in that city. Later he attended Toronto University, taking an engineering course. He spent some time prospecting in Northern Ontario and was with several surveying parties. Previous to his enlisting at Guelph he was assistant city engineer at Regina, Sask., to City Engineer Frank McArthur, now city engineer of Guelph.

THOMAS JOHN KENNEDY, president and general manager of the Algoma Central & Hudson Bay Railway Co., and the Algoma Eastern Railway Co., died at his home in Sault Ste. Marie, Ont., on August 30th, after an illness of several months' duration. Mr. Kennedy, who was in his sixty-second year, had spent most of his life in railroad circles, and was widely known in that connection. Early in life he was connected with the C.P.R. and was at one time superintendent of the North Bay and Chapleau branch. In 1900 he became superintendent of the A.C.R. and A.E.R. In 1901 he constructed the extension of the Algoma Eastern from Cream Hill to White Fish, and at the time of his death was one of the receivers of the A.C.R., in company with Mr. Vivian Harcourt, of Montreal.