PREDICTION OF ARMOURED FACE CONVEYOR PERFORMANCE FOR INCREASED EFFICIENCY

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ABSTRACT

The installed power and size of Armoured Face Conveyors are increasing along with the production requirements from longwall coal faces. The usual simple methods of calculation tend to overestimate the amount of power needed to run a conveyor. A computer model has been developed in the University of Newcastle Upon Tyne for the dual purpose of analysing data from existing installations and for predicting the power needed for new applications.

The paper also includes engineering developments that are being used to improve conveyor life and performance.

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1. INTRODUCTION

In the present difficult economic climate worldwide, great pressure is being placed on the coal mining industry to reduce costs and improve productivity. In U.K. underground coal mines, where most of the coal is mined from longwall faces, the trend is for fewer more productive units. This has resulted in an overall increase in face length. As face length increases, the power needed by the Armoured Face Conveyor (A.F.C.) increases. This is because more chain assembly and coal load has to be moved with an increase in conveyor length. However, the stage loader and shearer power requirements are relatively unaffected by increases in face length.

With the increase in power, larger chain sizes are needed to preserve the factor of safety of the chain assembly, particularly when starting up a fully loaded conveyor. The increase in conveying capacity and power requirements in recent years is clearly demonstrated by the increases in chain sizes in the last ten years. Chain sizes have increased from 18mm to 22mm and 26mm in diameter; some manufacturers are even contemplating designs using 38mm and 42mm diameter chains.

This trend cannot continue indefinitely as It becomes more difficult to handle the heavier components underground. Therefore, recent research has been focused on methods of predicting power and analysing conveyor performance in an effort to make recommendations for improving conveyor performance and efficiency. It is hoped to reduce the power consumption, and to increase the carrying capacity. **The** data used in this work was obtained from measurements taken on conveyors being tested on surface and underground. Surface testing was carried out at the British Coal test site at Swadlincote and at the Mining Supplies Longwall Ltd. test site in Doncaster. Some of the conveyors tested on surface were later tested underground after they had been installed at a colliery. This enabled the comparison of results from surface and underground testing.

A more systematic engineering approach had to be employed if reductions in installed power or improvements in conveying efficiency were to be made. First a thorough understanding of the mechanical theory describing the interaction of the component parts of an A.F.C. was developed. Then models could be made of thp effect of changes in the parameters governing the power consumption and carrying capacity of A.F.C.s.

Mathematical models and methods for calculating the drive power of A.F.C.s exist. For example, those described by (Bates,(1969), Brook (1971), Linke (1976), Schaefer (1970), and Société Stephanoise (1979). These methods of calculation have been derived from a similar method used to calculate the power needed to run belt conveyors. Where the total mechanical power needed is the sum of: the power needed to move the empty conveyor, the power needed to move the coal load and the power added or subtracted due to changes in the gradient.

These methods of calculation find values for power that are directly proportional to coal load. This is because the only variable value that changes is coal load; all other variables are assumed to be constants. It was found that these models do not calculate the power required sufficiently accurately when large coal loads (over 800 tonnes/hour) are being conveyed, or when large pre-tensions are applied. The solution to the inaccuracy of these calculations was to use empirically found factors to adjust the value found for installed power. In this way the calculations could predict with acceptable accuracy the power required for new conveyors of similar configuration in similar seam conditions. However, once the seam conditions changed or some operational characteristics were altered the predicted power was an unacceptably incorrect estimate of installed power.

The reason for the inaccuracy of the estimates is that the basic method of calculating power does not take into account variable losses such as those -arising from snaking, pre-tension and frictional losses around the headframe.

2. MODELLING THE A.F.C.

The mathematical model of the A.F.C. was developed from the most simple system of forces occurring in the steady state, and by clearly stating the assumptions made. The initial assumptions made were then relaxed as the model was improved. Assumptions had to be made so that the mathematics did not become unreasonably or unnecessarily complicated. During the work it quickly became apparent that the theoretical model of an A.F.C. cannot fully describe all the phenomena influencing power that occur in a real A.F.C. in every detail, no matter how complicated the model. But by accounting for more variables and the interactions that occur between them) during the operation of an A.F.C, it has been possible to reduce the difference between theoretical power and actual measured power.

The electrical power input to an A.F.C. is required to overcome the numerous losses in the machine and do useful work. Various frictional phenomena create resistances to motion which, in turn, create chain tension. It is the chain tension opposing motion that needs to be overcome for the conveyor to transport coal. The increase in chain tension may be created by direct frictional effects (of one body moving against another) or by mechanical effects such as the snaking of the conveyor. Under steady state conditions the increase in chain tension is created by the following:

Frictional effects.

a) Coefficients of resistance.
b) Chain link interaction, particularly at drive sprockets.
c) Internal resistance of the coal load.
d) Headframe and snaking frictional losses.
e) Conveyor gradient.

Mechanical effects.

f) Chain extension due to snaking.
g) Pre-tension.
h) Changes in sprocket pitch.
i) Manufacturing tolerances.
j) Horizontal and vertical changes in alignment.
k) Headframe angle and discharge type.

The variables mentioned above create chain tension that opposes the motion of the A.F.C. drives. Some of the effects have been found to be more important than others.

The amount of simplification of a problem and the quality of the data determines the accuracy of any result, provided that the influence of the most important (critical) variables have been included in the model. The most critical variables in the calculations were identified using sensitivity analysis on a model based on the calculations of Bates (1969), and Schaefer (1970). The most critical variables were found to be the various coefficients of resistance occurring during operation, the coal load mass per metre and the pre-tension.

Other variables, e.g. transmission losses, were found to have less effect on the value of power as their value was small compared to the total drive power. In the experimental work great care was taken to ensure that coal load, pre-tension, gradient and all other measurable data were recorded accurately.

The mathematical model developed to describe the A.F.C. clarified the processes that create the various resistances to motion in the steady state and how they absorb the power supplied to the A.F.C. The calculations take into account the effect of coal load, carryback, point of loading, cleaning load, gradient, pre-tension, snaking and headframe friction. The effect of tension changes due to snaking were considered as a variable part of the applied pre-frension in the chain assembly. The braking effect exerted on the chain assembly, as it passes through the various undulations in the pan line and around the headframe, was included in the dynamically created effects that cause further resistances to motion.

All calculations using the model are performed with the aim of finding the tension in the chain assembly at various points around the conveyor, particularly at either side of the drive sprockets. Figure 1 shows a simplified scheme of the forces acting in an A.F.C*

* Nomenclature given in Appendix.

Return drive

Discharge drive

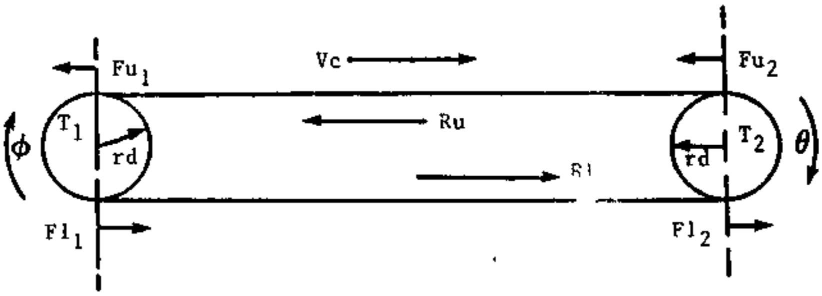

Figure 1. Forces acting

For moticn to oon" e fof? M drive sprockets to the chain assembly T ^ 1 <• e^n ü *** he sura of the resistances to** iaoti in the t< **e** *i* **the conveyor. This is** ***described*** **ъ> •• mc**

$$Fu\hat{} \quad Fu\ i = Ru \tag{1}$$

$$Fl_1 \quad Fl- \tag{2}$$

The difference between chain force (tension) on **the incoming side** of the drive sprockets must be equal and opposite to **the** drive **torque** as described by equations (3) and (4).

$$Fl \quad Fun \quad \frac{T_2}{rd} \tag{3}$$

$$Fu, \quad Fl \quad - \frac{T_1}{rd} \tag{4}$$

Equation 5 may be derived to show the relationship between **the** minor chain tensions in the chain assembly on the run off side of **the** drives, load resistance and the available motor torques.

$$Fu_1 - Fl_2 = Rl - \frac{T_1}{rd} = - Ru + \frac{T_2}{rd} \tag{5}$$

Because the applied torque at the drives is required to overcome the steady state load resistance to motion, the sum of the applied torques must be equal to the sum of the load resistances in the **steady** state. This is described by equation (6).

$$\frac{T_1}{rd} + \frac{T_2}{rd} = Rl + Ru \tag{6}$$

It is assumed that when the sum of equation (5) becomes zero all slack chain produced will occur on the run off **side of the return drive.** This difference in force may be calculated in terms of the extension of the chain assembly. Equations describing **the** minor forces Fu and Fl are used to calculate force taking into account **the** static chain tension (pre-tension) and that created dynamically by the resistances to motion. The major chain forces Fu and Fl are found by calculating the total resistance to motion in each race. The mechanical power needed at each drive sprocket is found using equations (7) and (8).

$$Pr = \frac{Zi}{rd} * \theta = (Flj - Fdj) * Vc \qquad (7)$$

$$Pd = \frac{Il}{rd} * 9 * (Fu_2 - Fi_2) * vc \qquad (8)$$

It is assumed that the angular velocity of the drive sprockets and the linear velocity of the chain assembly *are* the same as there can be no slippage of the chain at the sprockets.

3. ANALYSIS AND PREDICTION

Knowledge of the chain forces and their distribution in the chain assembly of an A.F.C. is required if existing A.F.C. designs are to be made more efficient. The magnitude of the chain forces occurring during the conveying process are difficult to measure due to a lack of suitable instrumentation. However, data obtained during performance testing of conveyors is available. It is this data that was analysed using **a** computer program developed at the University of Newcastle Upon Tyne. The program enables the chain forces occurring during the conveying process to be calculated.

There is great difficulty in obtaining by experimental means good values for coefficients of resistance. Therefore, the program was used to estimate values for these variables by back analysis. Some performance testing of conveyors was specifically carried out for this work to enable these estimates to be made.

The performance testing data was obtained from conveyors tested on surface and underground. Five main tests were carried out as listed below.

a) No-load, no snake or undulation, varying pre-tension.
b) No-load, with snake no undulation, varying pre-tenslon.
c) No-load, with snake and undulation, varying pre-tension.
d) Loaded, no snake, no undulation, constant pre-tension, varying coal load.
e) Loaded, with snake and undulation, constant pre-tension, varying coal load.

Some additional tests were carried out to measure the power consumption of unloaded drives.

The performance testing data recorded for the conveyors studied contained values for all the variables included In the computer model important to the calculations with the exception of the coefficients of resistance. The program was used to vary the coefficients until the theoretical mechanical power was close to the actual mechanical power measured in the tests. It was hoped that when the powers matched, the theoretical values for the coefficients would be close to their actual values.

Using the performance testing data, values found for the coal on steel coefficients of resistance, were between 0.38 and 0.52 for the conveyors tested on surface. The coefficients for steel on steel in empty conveyors were found to be between 0.17 and 0.25. Values found for conveyors tested underground were on average higher. The greatest coefficient was estimated at 0.58 for coal on steel. No values could be obtained for steel on steel as conveyors tested underground were always contaminated with coal to some extent.

The values found were on average higher than those found by Guder (1968) in similar experimental work, and lower than the co-efficients of friction measured by Evans and Pomeroy (1966). The reason for these differences when comparing the results is the difference in the qualities of British and German coals, and also the experimental procedures used to obtain the various results. Evans and Pomeroy describe how coefficients of friction vary with coal quality. It was also suggested that they may vary with coal strength.

As British coals are generally stronger than German coals, the fact that the values found for coal on coal coefficients of resistance in this work were higher than those found by Guder is not surprising. The values found for the steel on steel coefficients were very similar.

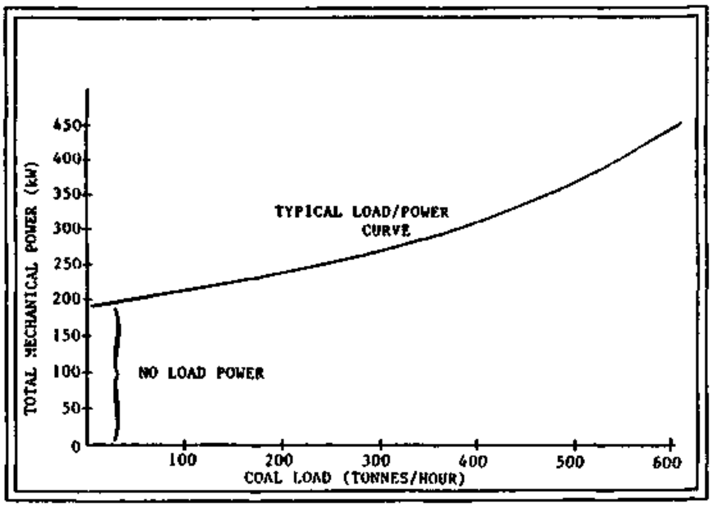

Figure 2. Illustration of typical results.

Figure 2 illustrates the typical trends obtained when analysing the data from conveyors tested.

The figure shows that the relationship between power and coal load is a curvilinear relationship as opposed to the linear relationship obtained using traditional methods of calculation. It was found **that a** linear relationship is acceptable up to 400/600 tonnes per hour. Above 600 tonnes per hour the graph becomes curvilinear and traditional methods of calculation are not as accurate.

The non-linear increase in power was shown to be due to increases in the values of coefficients of resistance. In particular, due to the braking effect at the headframe and in the snake caused by increasing chain tension.

4. ENGINEERING DEVELOPMENTS

Over the past ten years the maximum production capacity of A.F.C.s has risen from approximately 400 - 600 tonnes per hour, to present peak production capacities in excess of 2500 tonnes per hour.

If meaningful power savings are to be made, it is 'the overall chain force (when running) that should be reduced. This may be achieved by making changes in the materials used, the conveyor **design** and the mode of operation.

One method of increasing the discharge rate (load carrying capacity/hour) is simply to increase **the** chain assembly velocity. The disadvantage of this is the increased wear rate of the pan **line.** An alternative approach to obtaining higher capacities is to **run the** conveyor more slowly. This has the disadvantage of increasing the chain forces due to the higher frictional resistances created. **This** necessitates the use of larger chain sizes to ensure a suitable factor of safety is maintained.

Pre-tension and snaking are detrimental to conveyor performance. Snaking should be avoided; where possible the snake should **be gentle** and located in the unloaded section of the conveyor. Some method of pre-tension adjustment should be used if it is available, such as that described by Wiechers (1986). As conveyors increase in **length** and ever increasing coal loads are carried, the ability to control pre-tension will become more important.

Side Discharge or Cross-frame Side Discharge units should **be used where** possible to help reduce frictional resistances and carry-back.

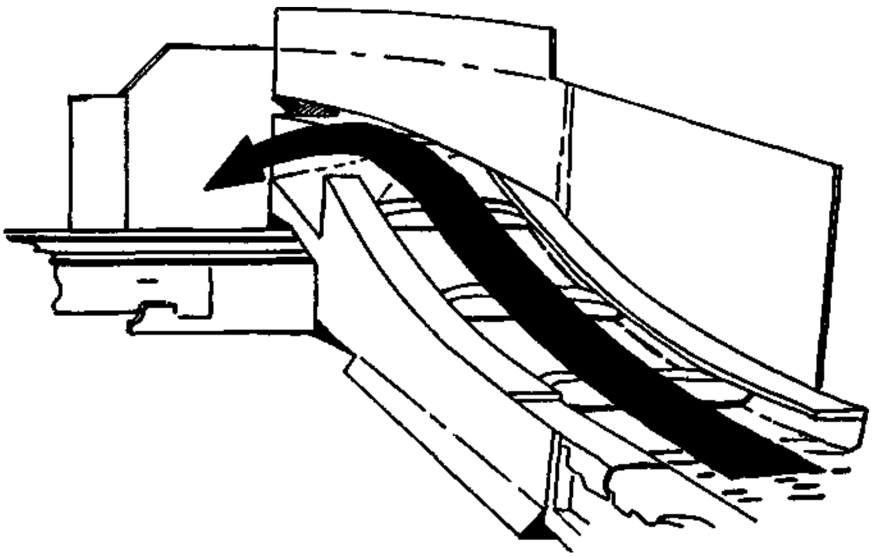

Figure 3. Side Discharge

Side Discharge drive frames reduce the power required by reducing the carry-back of coal into the bottom race. Cross-frame Side Discharge units reduce power by creating less carry-back and by lowering the braking resistance by using smaller ramp angles.

The use of friction and wear reducing materials have been shown to reduce power and increase conveyor life when used on components such as flight bars, ramp guides and pan lines.

5. CONCLUSIONS

Detailed analysis of the components making up the A.F.C. has provided a computer model which agrees with test results obtained from surface and underground installations. The theoretical results obtained using the model offer more accurate predictions than those obtained from previous methods of calculation.

This work has shown that further investigations into the influence of coal properties, the use of materials other than steel for flight bars and pans would yield benefits In terms of Improved life and reduced power consumption.

6. ACKNOWLEDGEMENTS

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APPENDIX

Glossary of Terms

Pre-tension

The static tension applied to the chain assembly.

Snaking

Snaking is the name given to the articulation of the pan line in the horizontal plane. Snaking creates additional tension In the chain both statically and dynamically.

Headframe

The headframe is the ramp at the discharge end of the face conveyor. It creates additional frictional losses due to the braking effect caused by the tension in the chain assembly and the ramp angle.

Carry-back load

Carry-back load is that part of the coal load carried back **into** the bottom race of the conveyor. The amount of carry-back has been found to be a function of discharge type, chain assembly velocity, conveyor load and the type of coal and its condition.

Coefficients of resistance

The coefficient of resistance occuring at a point in the pan line is the equivalent coefficient of friction that takes into account the various frictional effpcts occuring at that point.

Nomenclature

Fl_2	=	Minor chain tension return drive	kN
Fuj^	=	Minor chain tension discharge drive	kN
F1l	=	Major chain tension return drive	kN
Fu_2	=	Major chain tension discharge drive	kN
Pd	=	Mechanical drive power discharge drive	kW
Pr	=	Mechanical drive power return drive	kW
rd	=	Drive sprocket radius	m
Rl	=	Total frictional resistance lower race	N
Ru	=	Total frictional resistance upper race	N
T2	=	Applied torque discharge drive	Nm
Tl	=	Applied torque return drive	Nm

V_c	=	Linear chain assembly velocity	m/s
$\dot{\theta}$	=	Angular velocity discharge drive	rads/s
$\dot{\phi}$	=	Angular velocity return drive	rads/s